# STEAM POWER PLANTS

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*ENGINEERING RECORD SERIES*

# STEAM POWER PLANTS

## THEIR DESIGN AND CONSTRUCTION

BY

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*Consulting and Mechanical Engineer*

*THIRD EDITION*

Entirely Rewritten and Enlarged

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# INTRODUCTORY NOTE

(TO THE FIRST EDITION)

FREQUENTLY engineers and others in charge of a manufacturing business, be it a mill, factory or electric generating station, are called upon to design and purchase a steam power plant or parts of it when their knowledge of the machinery that goes into such a plant is more or less limited, and without being able to obtain the benefit of the advice of a competent consulting engineer. It is hoped that this book will be of special value to this class and of some value to all interested in steam power-plant construction. Part of the text appeared in a series of articles in THE ENGINEERING RECORD, and when the demand for them seemed to warrant their being published in book form they were thoroughly revised and considerable new matter added. A number of the illustrations have been selected from articles printed in THE ENGINEERING RECORD during the last two or three years descriptive of steam power-plant construction. They are reprinted without the text that accompanied them, thinking they would be suggestive.

# INTRODUCTORY NOTE

(TO THE THIRD EDITION)

SHORTLY after this book was written in 1902, the author began to practice as a consulting engineer for power-plant design and construction, and, upon revising the work for the edition appearing in 1912, the experience gained has led to some changes and many additions. The chapter on turbines has been added, also much new matter upon the subject of steam piping, condensers, and chimneys. Other parts of the book have been added to and brought up-to-date and a number of new illustrations are presented.

H. C. M.

NEW YORK, *January, 1912.*

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# STEAM POWER PLANTS.

## CHAPTER I.

### THE DESIGN OF STEAM POWER PLANTS.

No better service can be done the non-expert about to construct a steam plant than to advise him to engage at the outset of the project some capable engineer to design the plant and superintend its installation. In spite of the advantages of having work planned and carried out by such men, it will always be, probably, that a very considerable proportion of the work done will be constructed, for one reason or another, by persons with a semitechnical training without the aid of the expert engineer. It is, therefore, proposed to give data and information which, it is hoped, will aid the engineer who cannot be called an expert, in selecting the various kinds of machinery and apparatus that make up a steam power plant; to discuss specifications for these in a general way; and, in some instances, to outline the manner in which they should be purchased.

It is the practice of many engineers in steam-plant construction to invite bids on apparatus described very generally in a specification and intended to perform a service under conditions that are named, the idea of the engineer being to allow each bidder to proportion the parts of the apparatus he is to furnish and to quote a price on it. When bids are received under these conditions, it generally follows that there is a variation in the size of the machinery offered by different makers to do the same work, and the lowest in price may not be best adapted for the conditions. An engine, boiler, or pump, in fact almost everything about a steam plant, may do the work required of it, but it may be so proportioned as to do it in a manner that is not best for the owner. For instance, while an engine may give the required power, its cylinders may be so small that it requires an excessively large amount of steam to run it, or a boiler may be so small that an abnormal

amount of coal must be burned in order to generate the steam required. The expert engineer is, of course, able to detect and reject bids on deficient apparatus, yet when the size of the apparatus is fixed by the contractor it may happen that the purchaser, who is not an expert, will accept machinery which is not best adapted for the service that it is to perform, particularly if the too frequent custom of purchasing the apparatus lowest in price is followed.

There are other methods of buying apparatus for a steam plant. One is to go to a reputable manufacturer or contracting engineer and engage him to build the machinery wanted and pay what is asked for it. Such a contractor should not, however, be placed in competition with others if he is to design a plant to fill the requirements of the owner; for if this is done the contractor's interest is to design a cheaper plant than will be proposed by his competitor, whose bid is based on the plant designed by him, and this kind of competition sometimes results in inferior apparatus being supplied.

Another method of purchasing is to have the engineer state in his specification the dimensions of the apparatus wanted, permitting, however, a departure from these specified dimensions in order that a manufacturer can make use of his standard patterns, provided such a course is not detrimental to the purchaser's interest. When all manufacturers bid on the same basis their bids are lower and the purchaser is sure of getting a properly proportioned machine, provided the engineer who prepares the plans and specifications is capable. Many believe the last two methods of procedure much preferable to the first. Latitude given an imprudent contractor, in the way of fixing the dimensions of the apparatus he is to supply, may, in a measure, be covered by demanding a guarantee as to efficiency; yet tests necessary to determine if the guarantees are fulfilled are expensive and are therefore generally omitted.

**Location of Plant.** — One of the first questions to be decided in the construction of a power plant is its location. This depends on several factors, the most important of which are, the ease with which the power may be transmitted from the generating source to the locations of the demand, the cost of delivering coal and removing ashes from the power house, and the availability of a supply of water for condensing purposes.

The first factor, the ease with which power may be transmitted from the generating source to the machines utilizing it, is the one that frequently decides whether an electric or belt-and-shafting system of transmission is to be used. It is, of course, impossible to give any general rule applicable to all cases, as each situation demands a thorough investigation of the cost of installing and operating both plants. In arriving at their relative costs of operation, the interest on the investments, the repairs, the fuel cost, the cost of attendance, supplies, etc., have to be reckoned. Generally speaking, it may be said that in any situation where the load on the power plant is practically constant throughout the working day, as in a textile mill, and where the power house may be located close to the lines of shafting so that belts may be carried from the engine to the shafting without the use of gearing, quarter-turn belts, etc., the belt-and-shafting system of transmission seems the most favored. If, however, the manufacturing establishment consists of a number of separate buildings in which lines of shafting are at different angles, so as to require several separate power plants or else a complicated system of belts or gears, the electric system possesses advantages as regards cost of operation that make its adoption advisable. Again, in establishments where the work is similar to that of a large machine or bridge shop, where the tools are used intermittently, where one or two departments may run overtime, the electric system is rapidly gaining friends. The reason for this is not the difference in the operating expenses, which are slight, inasmuch as a machine shop requires a very small amount of power for its operation, but in the greater convenience and cleanliness of the electric system. One solution of the problem of transmitting power when shafts lie in directions that are not parallel is the rope drive. This has been used with the greatest success in many plants. Its low first cost and cost of repairs and the flexibility of the system make it well worth considering in many installations.

The cost of handling coal and removing ashes in a power house may be a considerable item in the operating expenses, and large plants are therefore usually equipped with coal-handling machinery. Some steam plants are so arranged that railway cars or coal cars may be run over a receiving hopper from which the coal is conveyed mechanically to storage bunkers over the boilers and

chuted from the bunkers to the furnaces by gravity. If a plant is not to be provided with coal-handling machinery, it is well, if convenient, to provide a trestle so that cars may be run over and dump into a bunker opposite the furnace doors in order that the coal may fall by gravity through holes in the wall separating the bunker from the boiler room, onto the floor of the latter, in front of the furnace doors.

The value of water in sufficient quantity to condense the steam exhausted by the engines often determines the location of the power house. From 15 to 20 per cent of the fuel used by a non-condensing engine will be saved if it is operated with a condenser. Each pound of steam exhausted by an engine requires a supply of 30 to 35 pounds of water for the condenser, so it will be seen that the water needed for condensing purposes is often a considerable quantity. Sometimes when a power house cannot be located on the bank of a stream from which a supply for this purpose is available, a pipe or conduit can be laid from a river to a well near the power house, the grade of the conduit being below that of minimum low water. The injection pipe, as the pipe that conveys the water to the condenser is called, can then be run from this well to the condenser, which is usually located close to the engine. Because of the vacuum in the condenser, the water will rise in the injection pipe from a lower level to the condenser. It is not advisable, however, to attempt to lift the injection water over 20 feet. Where the lift would be slightly greater than this the condenser can be placed in a pit. Since the development of the cooling tower, an apparatus for cooling condensing water so that it can be used over and over again, condensing plants are not so dependent on an abundant water supply as they were before this apparatus was perfected; hence the importance of locating a plant by a stream for condensing purposes is not so great as it was once.

**Drawings.** — Complete and accurate drawings of all of the details of a steam plant are a necessity. It is well to make assembled drawings showing the plant in plan and as many elevations as may be necessary to make the arrangement perfectly clear. Assembled drawings insure all parts of the plant fitting together properly, and prevent mistakes such as attempting to run steam pipes where a building column ought to be, and so on. If assembled drawings are to be made, the scale assumed should be

sufficiently large to show the steam-pipe system. Three-eighths of an inch to the foot is about as small as can well be used. As soon as contracts are made for engine, boilers, feed-water heaters, pumps, etc., accurately dimensioned blue-prints showing the machinery in plan and elevation should be obtained from the contractors. Experience has shown that it is well, with some firms at least, in order to avoid delay, to get these blue-prints before the contracts are let. Contracts for the building, steel-work, etc., should not be made until the machinery is contracted for, since it may not be possible to obtain the machinery which was contemplated at the time the building plans were made, and machinery then available may not fit. It is, of course, essential that the building be fitted to the machinery, not the machinery fitted to the building.

**Type of Power House.** — Generally the relative location of the engine and boiler rooms is determined by some local condition. Where it can be done, however, it is better to locate the engine and boilers in one building, with a wall between them, and to place them in parallel rows with the cylinders of the engine adjacent to the rear of the boilers. This is particularly the best arrangement if the plant is likely to be enlarged in the future. The steam pipe connecting the boiler and engine or engines is the most direct in this arrangement, and it can be most readily enlarged. If the engine and boiler houses are placed end to end and contain two or more boilers supplying one engine, the proper size of piping for such a plant will be inadequate if another engine is added at one end of the plant and more boilers installed at the other.

Where land is very expensive, boilers are sometimes placed in buildings on two or more floors. Ordinarily, however, the boiler-room floor is usually on the level with the outside ground, while the engine-room floor with large engines is invariably higher, usually from 6 to 12 feet or more, depending on the height of the engine foundations. Sometimes an engine is installed where the engine-room floor is on the ground level. When this is done, it is necessary to construct a pit for the condenser, if such an auxiliary is used, also pipe trenches, and, if the engine is a large one, a pit for the flywheel. It perhaps ought to be stated at this time that a condenser ought always to be below the engine, as the pipe leading to the condenser cannot rise at any point without

introducing a dangerous element into the power plant. The arrangement that necessitates the construction of pits ought to be avoided, if it is possible, as it is difficult to construct a water-tight pit for the flywheel and condenser, because the vibration of the engine is apt to crack the lining of the pit and allow water to enter, if there is any in the soil. The trenches for the exhaust piping have to be covered and the piping is not nearly so accessible when in a covered trench as it is when in the basement usually provided under engine rooms.

**Building.** — A building for a power house should, if possible, be constructed of fireproof materials. Brick walls with steel trusses supporting a wooden roof covered with tar and gravel, or with some form of fireproof construction, is the usual construction. The brick walls sometimes carry the roof trusses and tracks for traveling cranes, and again these are supported by steel columns resting on the foundations and imbedded in the brick walls, the latter, however, carrying only their own weight. The building should be designed by an architect or structural engineer. The construction of the building can be done by bridge shops making a specialty of constructing buildings of this character and supplying the steelwork for them.

The buildings should be of such a size that the machinery in them is not cramped. When there are several machines in an engine room, it should be remembered when locating them that it is sometimes necessary to stop a machine very quickly. It is well, therefore, to place all machinery in such a position that it is readily accessible to the man or men in charge of it. Provision should be made, in planning and constructing a power house, for bringing the machinery into the building after it is erected, and a door of sufficient size to admit the largest part of a machine must be provided. Ample room must be left around the horizontal steam and water cylinders to be able to remove piston rods if it should be necessary, without removing the cylinder from its foundations. Enough space must be left between the foundations of adjacent engines and between foundations and engine-room walls to allow a man to get between them to reach the foundation bolts. In a boiler room, there must be a clear space in front of the boilers at least as wide as the boiler tubes are long. The distance between the rear of the boilers and the wall need not be greater than 5 or 6 feet, or enough to allow a wheelbarrow to

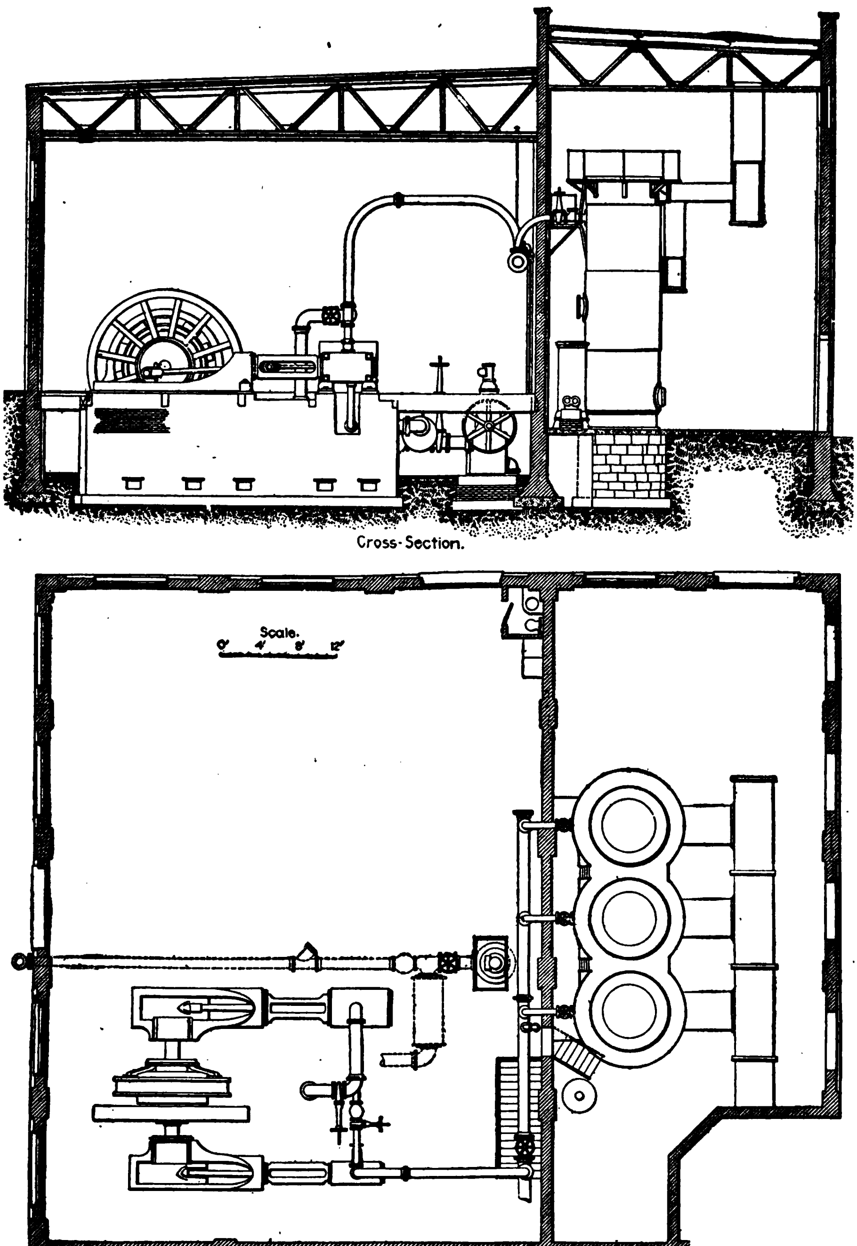



**Fig. 1.** Electric Power Station designed by Dean & Main.

be placed opposite the soot door in the rear of the boiler setting, if such a door is provided.

**Foundations.** — The character of the soil underlying the site for a power house should be carefully examined in order that the

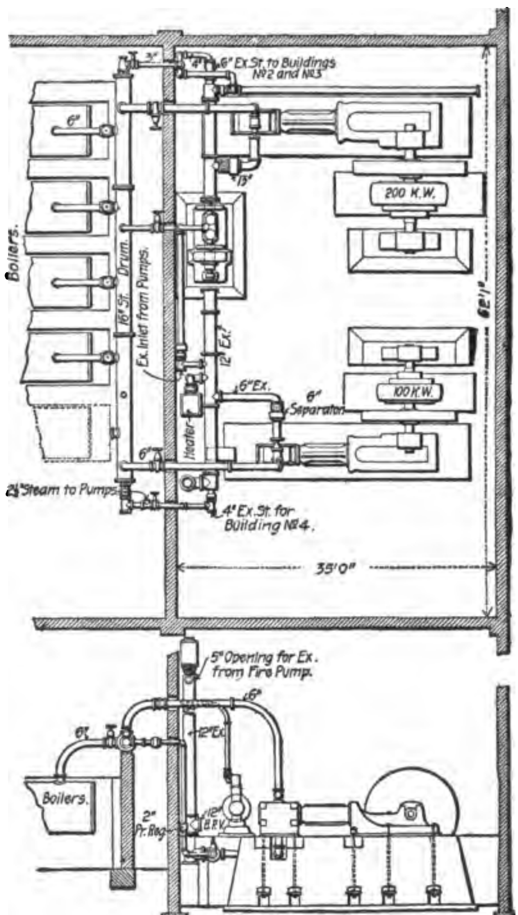



FIG.2

Fig. 2. Piping in Power House. (Lockwood, Greene & Co., Engineers.)

foundations can be so planned as to keep the load imposed by the buildings and machinery within the safe limit. For ordinary one-story buildings where the loads are not excessive, holes should be dug at numerous points over the site and the character of the soil

determined. As the magnitude of the work increases more care should be taken. In important work a competent specialist in foundation work should be consulted. As to the bearing capacity of different soils, the New York Building Code states: "Different soils, excluding mud, at the bottom of the footings shall be deemed to safely sustain the following loads to the superficial

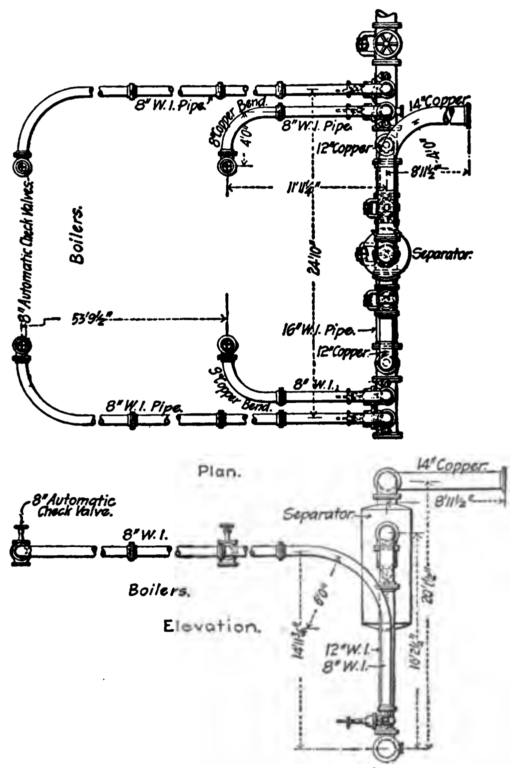



**Fig. 3. Section Main Steam Piping, Lincoln Wharf Station.**

foot, namely: Soft clay, one ton per square foot; ordinary clay and sand together, in layers, wet and springy, two tons per square foot; loam, clay, or fine sand, firm and dry, three tons per square foot; very firm, coarse sand, stiff gravel or hard clay, four tons per square foot."

If loose rock is found it should be removed; solid rock should be dressed off in steps with vertical risers and horizontal treads

so that the pressure will be exerted everywhere in a vertical direction. Solid rock will stand almost any load that can be imposed upon it. If soil of low bearing power is found, piling is usually resorted to. Piles may be of spruce or hemlock, at least 5 inches in diameter at the point and 10 inches in diameter at the butt for piles 20 feet or less in length, and 6 inches at the point and 12 inches at the butt for piles over 20 feet in length. The bearing power of piles not driven to rock or hardpan or similar firm material may be calculated by the Wellington formula, in which the safe bearing power in tons is equal to twice the weight of the hammer in tons multiplied by the height of the fall in feet divided by one plus the penetration of pile under the last blow in inches. From a knowledge of the total load to be carried and the load that each pile will support, the number of piles necessary under a weight to be supported can be calculated. If the soil is not firm, the bearing power of a pile should be taken much less than that given by the formula, in order to allow for decrease in strength due to vibrations of the machinery. To avoid decay the piles should be cut off at the level of the ground water or not more than one foot above it.

It is getting to be the practice in power-house construction where piles are used, to saw off the heads of the piles to a uniform grade, then excavate the materials between the heads of the piles for a depth of a foot or so and lay a bed of concrete sometimes several feet in thickness between the heads of the piles and over their tops. An 8-foot bed of concrete over piles on 30-inch centers was used as a foundation for the 96th Street power house of the Metropolitan Street Railway Company in New York City. If the soil underlying a power-house site is found to possess too low a bearing power for the foundations of the engines and boilers to be constructed directly on it, concrete beds may be laid so as to distribute the load. Sometimes a concrete bed is laid under the engine foundations and another under the boilers. Again, the entire site is covered with a bed of concrete and the wall footings and machinery foundations built directly upon it.

**Engine Foundations.** — These are almost invariably constructed by the owner, the engine builder furnishing the drawings. The latter generally consist of accurately dimensioned drawings showing the foundations in plan and one or two elevations, and also a drawing of a board template which has to be

made for locating the foundation bolts. A hole is bored in the template where each bolt is located, and the template is supported over the place where the foundation is to be constructed, at such a height that the foundation bolts may be suspended in the position that they will finally occupy, by passing them through the holes in the template, the nut on the upper end holding them in position. The foundation is then built around the bolts, leaving holes about them an inch or two greater in diameter than the bolts themselves, so that the latter may be moved slightly, to pass through the holes in the engine bed plate, should an error occur in locating the position of the bolts or the holes in the engine bed.

Foundations for engines are sometimes constructed on a very thick bed of concrete, as has been explained. This is not always necessary, and, where good loam and clay are known to exist for some distance below the surface of the ground, the earth may be leveled off and the foundation commenced on a layer of concrete just thick enough to give a good bearing. If loose rock or poor earth is found, this should be removed and the excavation filled with concrete, in which loose stones, old bricks, etc., may be imbedded, care being taken to put them in layers alternating with the concrete, which should be so placed and rammed as to make sure that there are no voids. This should be continued up to the level at which the regular brick or concrete foundations are to commence. There seems to be no good reason for using a pure concrete bed if the earth is of good quality and the load imposed upon it not excessive. Engine foundations should be of brick laid in cement mortar made of one part Portland cement to two parts clean sharp sand or of concrete. Good concrete is obtained by mixing one part good Portland cement, two parts sand, and four parts broken stone, the latter small enough to pass through a 2-inch ring. Concrete should be laid in layers not over 6 inches thick, each layer being thoroughly rammed before the one above is put down. Such a foundation has to be surrounded with board walls or forms to suitably hold the concrete while it is being laid. A Portland cement concrete foundation ought to stand at least two months, and one of brick in Portland cement one month before it is loaded, whenever possible to spare the time; but if suitably proportioned and carefully watched it can be safely loaded much sooner, especially if the concrete is made as dry as possible.

**Operating Expenses.** — The cost of operating a plant includes the cost of fuel and removing ashes if coal is used, the cost of water, oil, waste, attendance, the cost of repairs necessary to keep the equipment in running condition, and the fixed charges. Fixed charges include the interest on the money invested, insurance, taxes, and depreciation. An allowance of 2 per cent of the cost of apparatus will ordinarily be sufficient to cover ordinary repairs. The rate of interest will be from $4\frac{1}{2}$ to 6 per cent, depending upon the value of money to the owners. An allowance of 2 per cent ought to cover both insurance and taxes, although insurance is not always carried.

It is customary in establishing a sinking fund to assume, in fixing the rate of depreciation, that a certain percentage of the original cost of power-plant apparatus shall be set aside or charged off each year to cover depreciation, and that each amount so treated will draw interest so that the total of the amounts so set aside plus the interest return on each will aggregate a sum sufficient to renew the apparatus at the end of its estimated life. To fix the rate of depreciation upon power-plant apparatus means, therefore, the determination of its useful life, and this is a very difficult matter to determine in many cases. The life of such apparatus is determined by its quality, that is, whether it is the best of its kind or of inferior manufacture, by the care and attention given it, by the use to which it is subjected, that is, whether it is run hard or moderately. In many cases the life depends mainly upon whether the equipment will become sufficiently obsolete as to require replacement before it is worn out. The depreciation in large central electric supply stations was very great in the several large cities in the United States during the period from 1895 to 1905. At the beginning of that period several cities were supplied with electricity by direct-current stations operating at low voltage, which required the use of a comparatively large number of stations scattered over the area supplied. The development of alternating-current electrical machinery, in large units, made possible a great reduction in operating expenses; and this fact, coupled with the growth of the business, caused the abandonment of many direct-current stations after only a few years' use and the installation of new generating equipment in one or at most comparatively few very large stations supplying the entire area with alternating current

transformed into direct current by suitable converting apparatus at points where direct current was formerly generated. In fact, all electric supply stations are subject to high rates of depreciation on account of the generally rapid increase in business which requires the abandonment of plants for the more economically operated larger ones. In certain classes of work, as in mills and factories, where this condition does not hold, and in office buildings, which are rarely increased in size, power-plant apparatus may enjoy a much longer period of usefulness. Under the latter condition the life of power-plant apparatus may be considered to lie within the limits given in Table 1, depending upon the quality of apparatus used, the care given it, and the manner in which it is used.

TABLE 1. — LIFE OF POWER-PLANT APPARATUS.

| | Years. |
|---|---|
| Buildings, masonry | 40–50 |
| Chimneys, masonry | 40–50 |
| Chimneys, iron | 10–20 |
| Boilers, water-tube | 20–30 |
| Boilers, fire-tube | 15–20 |
| Engines, Corliss slow-speed | 20–30 |
| Engines, medium-speed | 15–20. |
| Engines, high-speed | 10–15 |
| Turbines | 20–25 |
| Pumps and condensers | 15–20 |
| Coal conveyors | 10–15 |
| Piping | 10–20 |
| Electrical generators | 15–25 |

TABLE 2. — RATE OF DEPRECIATION.

| Life in years. | Rate of interest, per cent. | | | | | | |
|---|---|---|---|---|---|---|---|
| | 3.0 | 3.5 | 4.0 | 4.5 | 5.0 | 5.5 | 6.0 |
| 5 | 18.83 | 18.65 | 18.46 | 18.28 | 18.10 | 17.91 | 17.73 |
| 10 | 8.72 | 8.52 | 8.33 | 8.14 | 7.95 | 7.76 | 7.58 |
| 15 | 5.37 | 5.18 | 4.99 | 4.81 | 4.63 | 4.46 | 4.29 |
| 20 | 3.72 | 3.53 | 3.36 | 3.19 | 3.02 | 2.87 | 2.71 |
| 25 | 2.74 | 2.56 | 2.40 | 2.24 | 2.09 | 1.95 | 1.82 |
| 30 | 2.10 | 1.93 | 1.78 | 1.64 | 1.50 | 1.38 | 1.26 |
| 40 | 1.32 | 1.18 | 1.05 | 0.93 | 0.83 | 0.73 | 0.64 |
| 50 | 0.88 | 0.76 | 0.65 | 0.56 | 0.42 | 0.40 | 0.34 |

Table 2 gives the rate of depreciation that should be charged for various periods of life of apparatus at various rates of interest return upon the sinking fund. For instance, if the assumed life is 20 years and the interest return on the sinking fund is 5 per cent, 3.02 per cent of the initial cost of the apparatus would be the annual depreciation.

**Cost of Power-plant Equipment.** — The author has been urged to furnish data as to the cost of power-plant apparatus, and has reached the conclusion, after an attempt to do so, that such information as might be given would be misleading and therefore had better be omitted, as the cost varies so with the quality and size of apparatus purchased, the location of the plan, and conditions governing its installation. It is far preferable for an engineer to obtain actual estimates of the various kinds of apparatus under consideration, and then all uncertainty will be eliminated. It is equally difficult to give costs of labor for power-plant operation.

# CHAPTER II.

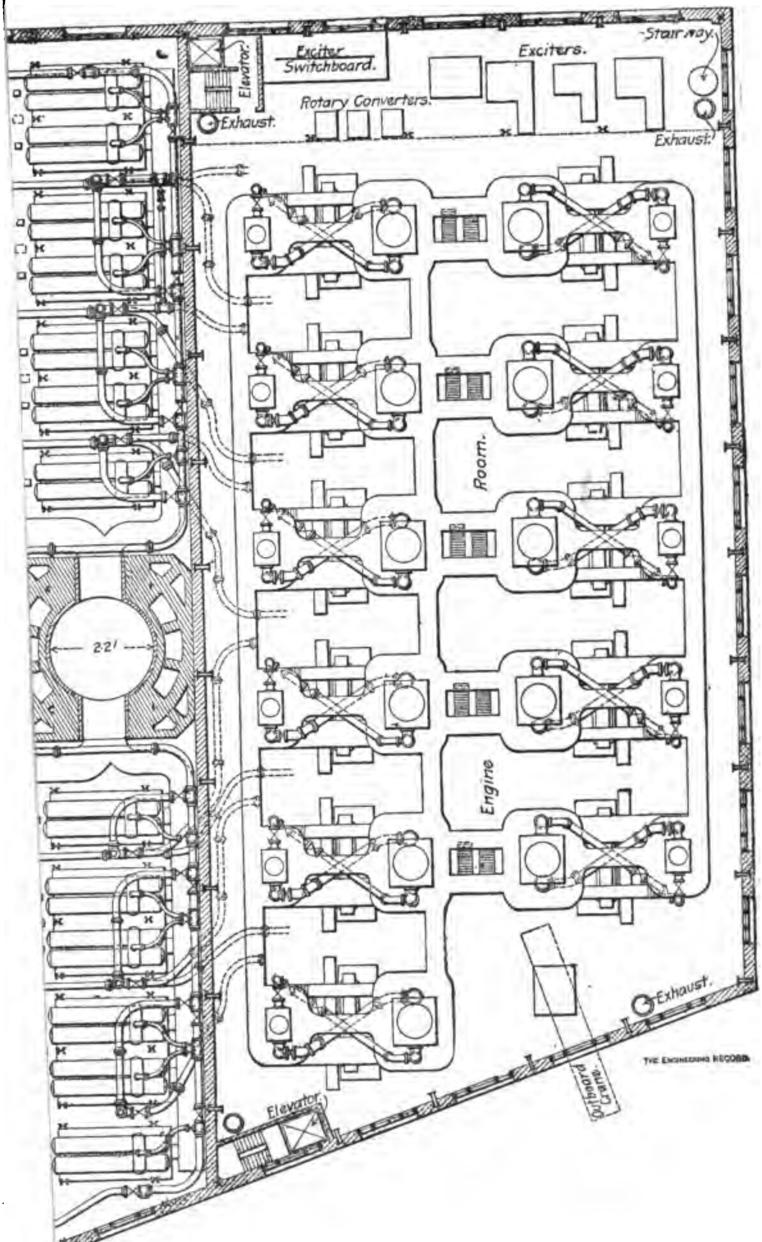
## PROPORTIONING STEAM BOILERS.

**Heating Surface Necessary.** — The function of a steam boiler is to transmit to the water it contains as much of the heat generated by the combustion of fuel as possible. Each square foot of heating surface in the boiler can transmit only a certain amount of heat when the highest economy is being realized. By increasing the supply of heat a greater amount is transmitted and consequently a greater amount of water is evaporated by each square foot of heating surface; but with the increase, the same percentage of the heat generated is not utilized. The reason is that, after a certain rate of evaporation is reached, the maximum capacity of the water to absorb heat is more nearly reached, and, hence, a larger percentage of the heat of the fuel passes up the chimney. In other words, it is possible to evaporate a certain amount, say 3 pounds, of water per square foot of heating surface in a given time and to utilize a certain percentage, say 80 per cent, of the heat in the fuel, 20 per cent going to waste. It is also possible to burn more coal and evaporate say 5 pounds of water per square foot of heating surface in the same time, but when doing so a greater percentage, perhaps 25 per cent, of the heat of the fuel is lost. The selection of the proper amount of heating surface for a steam boiler is, therefore, a very important matter.

The best efficiency, under ordinary working conditions, with most boilers, is obtained when evaporating about 3 pounds of water per square foot of heating surface per hour, from a feed temperature of 212° F. into steam at atmospheric pressure. This is equivalent to allowing nearly 12 square feet of heating surface per boiler horse-power. Most water-tube boilers are rated upon the basis of 10 square feet of heating surface per horse-power. Most boilers, sometimes, and quite frequently in fact, attain a very high efficiency when the rate of evaporation is considerably higher than that given. Such high results are

usually attained, however, when all of the many conditions affecting the efficiency are such as to produce a good result. Many of these conditions are not so favorable when a boiler is operated in ordinary service. For instance, if the boiler surfaces are not clean, owing perhaps to the accumulation of scale or soot, the efficiency of the heating surface will be more or less impaired. For this and other reasons, the writer believes it is well, at least in plants where the load is fairly uniform, to provide ample heating surface for the work to be done; for not only will such a course result in a saving of fuel at ordinary rates of evaporation, but it will make it possible to run a boiler considerably above its rating and still maintain a high efficiency. With electric generating stations or other situations where the maximum load is of short duration, so liberal an allowance of boiler capacity is not good practice; for some types of water-tube boilers can be driven at double the rating if there is sufficient draft to burn the necessary coal on the grates; and this can be done, too, without seriously affecting the efficiency. For this reason the saving in first cost may, in instances of this kind, more than offset the decreased economy due to driving the boilers above the normal rating for the short period of heavy load. Dr. D. S. Jacobus, in the Journal of the Franklin Institute for December, 1910, writes as follows concerning the rating of boilers in central-station practice:

"Much depends on the load curve of a power plant in obtaining economy. If a continuous uniform load could be carried, many of the vexing problems which confront the power-plant engineer would be eliminated. It is difficult to carry economically enough reserve capacity to meet the daily peaks in the load. Then, again, there are exceptional peaks which occur only at rare intervals, so that a considerable percentage of the available power may be developed only for a few hours every month, or, for that matter, for a few hours every year. Modern practice leads more and more to developing higher ratings from boilers during such intervals, and a boiler should be used which, under proper operating conditions, may be driven to a capacity that is limited only by the amount of coal which can be burned in the furnace. Again, it is desirable to use boilers that may be cut into the line quickly either from banked fires or starting from a cold state.




COMPANY, NEW YORK.

"The practice in this respect is exemplified by considering the installations of the Commonwealth Edison Company at Chicago, where the first 5000-kw. turbines erected in this country were installed. This was in 1903, and eight boilers each having about 5000 square feet of heating surface were supplied for running a turbine. The maximum rating for these turbines was 7500 kw. Later on 12,000-kw. maximum-rating turbines were installed, each with eight boilers of the same size as provided for the 5000-kw. machines. Still later machines of 14,000 kw. maximum were run with the same size and number of boilers as the original machines of 7500 kw. maximum."

**Value of a Boiler Horse-power.** — According to the American Society of Mechanical Engineers' standard, a boiler to develop 1 horse-power must raise 30 pounds of water per hour from a temperature of 100° F. to the temperature of steam at 70 pounds pressure, and evaporate it into steam at the pressure. This is equivalent to evaporating 34½ pounds of water per hour at a temperature of 212 degrees into steam at atmospheric pressure, or "from and at 212 degrees," as it is sometimes called. The term horse-power is frequently used when estimating the capacity of steam boilers, and the custom of buyers was formerly to ask for boilers of a certain horse-power. This is an improper way to buy boilers unless the amount of heating surface per horse-power is closely examined. If bids are called for boilers of a given horse-power, one bidder might offer a boiler with ample heating surface, while another might offer a boiler with much less. Both might develop the required horse-power, but the one with deficient heating surface might do it only at an increased cost for fuel, as explained in a previous paragraph. In saying this it should be borne in mind that the efficiency of various classes of surfaces varies according to their location and arrangement, but this variation in first-class boilers is confined to narrow limits which can only be detected by the expert.

**Advantages of Types of Boilers.**—Barrus, in his excellent work on "Boiler Tests," states that "the economy with which different types of boilers operate depends more upon their proportions and the conditions under which they work than upon their types; and, moreover, that when these proportions are suitably carried out and when the conditions are favorable, the different types of boilers give substantially the same result." So much for the side

of efficiency. As to safety, the water-tube boiler is generally considered to be far superior to boilers of the fire-tube type. While water-tube boilers seldom explode, although the tubes sometimes burst and injure those firing them, it is not believed that water-tube boilers as a class are immune from series explosions. Some types, however, have never experienced such a disaster. Water-tube boilers also possess an advantage in that

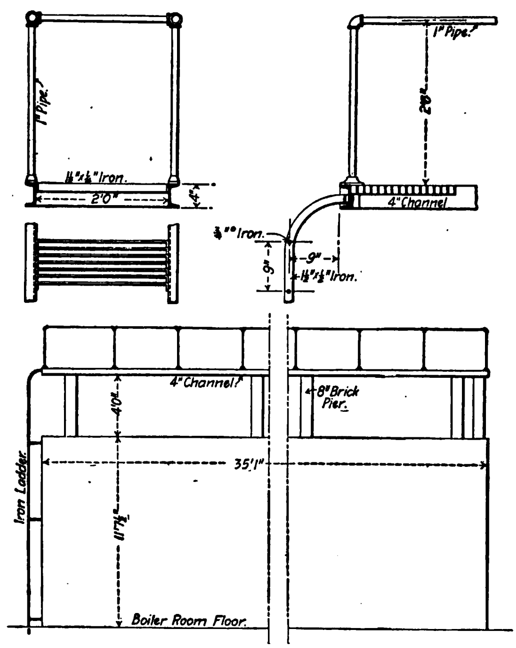



Fig. 4. Iron Walk over Boilers designed by Sheaff & Jaasted.

they contain less water than the shell type, as usually designed, and consequently steam can be raised in them more quickly. More water can probably be evaporated per square foot of heating surface in a water-tube boiler than in one having fire tubes, on account of a larger and longer passage for the gases, and possibly on account of a better circulation of water in contact with the heating surfaces. The efficiency of the two types is about the same, the difference depending more upon the proportions and management, as Mr. Barrus states, than upon the types.

Water-tube boilers have increased in favor very rapidly in the past few years, particularly for electric work, where large amounts of steam are wanted suddenly, and sometimes without previous warning. In spite of the fact that the water-tube boiler possesses advantages over it, the fire-tube boiler, particularly the horizontal return tubular boiler, is still used to a very considerable extent on account of its lower cost. An internally fired boiler, such as the Manning, locomotive, Galloway, or Lancashire boiler, all of which are of the fire-tube type, possesses advantages over the brick-set boilers in that there is not the air leakage into the furnaces, reducing the efficiency, that is often found in the boilers which require a brick setting. Furthermore, they do not need the costly brickwork. Internally fired boilers are, on the other hand, unless provided with large furnaces, very poorly adapted for burning bituminous coals, which require large combustion chambers, in order that the flame from the burning coal cannot come in contact with the relatively cooler surfaces of the boiler and thus become cooled to such an extent that combustion ceases and the gases pass to the chimney unconsumed and consequently wasted. For this same reason, horizontal tubular boilers must be raised a good deal higher above the grate if bituminous coal is to be burned than if anthracite is used. Internally fired boilers are also poorly adapted to burning small sizes of anthracite coals, or any coals that burn at a low rate, on account of the restricted grate area that is found with most boilers of this type. It is held by some engineers that bituminous coal gives the best results if it is burned in a furnace entirely separate from the boilers, so that the radiant heat from the fires will heat the fire-brick lining, or checkerwork, which is sometimes introduced in the combustion chamber, to such an extent that the gases coming in contact with this highly heated brickwork are consumed before reaching the boiler. Horizontal return tubular boilers are seldom used for pressures as high as 150 pounds, and this fact has a good deal to do with the preference engineers have for water-tube boilers, where pressures of from 175 pounds to 200 pounds are common, particularly with turbines. Probably the greatest advantages of water-tube boilers is the economy of space their use brings about of the large size units in which they may be obtained, units of 2500 horse-power being manufactured. Economy of space is of such vital importance in large central

stations that water-tube boilers are invariably used in such plants in the United States.

**Division of Heating Surface in Units.** — The first problem connected with the design of a boiler plant is to determine accurately the maximum number of pounds of steam that will be used by the various engines, pumps, and other parts of the plant which will have to be supplied. As has been said, 1 square foot of heating surface should be allowed in shell boilers for every 3 pounds of water to be evaporated into steam from and at 212 degrees in an hour's time. With water-tube boilers the heating surface can be obtained by dividing the number of pounds of water to be evaporated into steam from and at 212 degrees per hour by 3.4. With this proportion and sufficient draft and grate surface to burn the necessary amount of fuel, a boiler can easily be forced $33\frac{1}{3}$ per cent over this capacity and maintain a good efficiency. Some boilers can do much better than this. In a test of Babcock and Wilcox marine boilers, by the U. S. Navy Department, for the battleship Wyoming, the efficiency was 74.3 per cent and 69.1 per cent with an evaporation of 3.88 pounds and 10.52 pounds of water per square foot of heating surface per hour respectively. It ought to be stated that the maximum evaporation of a boiler is limited mainly by the amount of coal which can be burned upon the grate. If the draft is sufficient, a good boiler can develop a horse-power upon one-half of the surface recommended. By dividing the total number of pounds of steam that are to be evaporated from and at 212 degrees per hour by 3 for fire-tube boilers and by 3.4 for water-tube boilers, the amount of heating surface for plants with a constant load may be obtained with sufficient accuracy.

The next step is the subdivision of this heating surface into the proper number of boilers. This is of considerable importance, for careful study may result in much saving in the first cost and in the cost of operation. For instance, if boiler capacity equivalent to evaporating 28,800 pounds of water from and at 212 degrees an hour is required, 9600 square feet of heating surface in fire-tube boilers will be needed. If each square foot of heating surface may be overloaded $33\frac{1}{3}$ per cent, it is evident that if the 9600 square feet were divided among four boilers, one boiler might be shut down for repairs or cleaning, which is frequently necessary, and the other three run at $33\frac{1}{3}$ per cent overload and

still evaporate 28,800 pounds of water per hour. If the total heating surface was divided into three boilers, each of 3200 square feet of heating surface, two might not be able to run the plant alone, so a fourth or spare boiler would have to be supplied. This would manifestly be a poor division of power, as the money spent on the spare boiler would represent so much capital lying idle most of the time. The frequency with which a boiler is shut down for repairs or cleaning depends upon the attention given it and the character of the feed water.

**Importance of Proper Grate Surfaces.** — To evaporate a given amount of water into steam, it is necessary to generate a certain amount of heat by the combustion of fuel. The factors controlling the amount of heat generated are, the kind of coal, the amount of grate surface the boiler contains, and the draft. Ample grate surface, therefore, is highly desirable, particularly when boilers are to be forced. The draft affects the rate of combustion, as the amount of coal burned on each square foot of grate surface in an hour's time is usually called. Of the factors mentioned, that pertaining to the kind of coal to be used should be determined by an engineer before designing a boiler plant. He should investigate the cost of the various fuels available at the locality at which the boiler plant is to be constructed, and, with these data and a knowledge of the relative evaporative power of the different fuels, he can determine in advance the cheapest kind of coal, the evaporative power and cost of each being considered, and construct the boilers so that it can be used. Many plants have been seriously handicapped by the failure of their designer to consider this subject. Some coals, besides having less heating power per pound than others, cannot be burned at as high a rate of combustion on account of peculiar properties they possess. For instance, with the finer sizes of anthracite coal, it is impossible, unless mechanical draft is used, to burn more than a small amount on a grate, as the particles pack together so closely that the air cannot get through the bed of fuel in sufficient quantity to permit a rapid combustion. Again, a coal high in ash and sulphur is limited in the rate at which it can be burned. Of course, if only a limited amount of fuel can be burned per square foot of grate surface in a given time, the grate has to be made large in proportion to the heating surface, and, when it is intended to burn a low-grade fuel, provision for a large grate

ought to be made in the first place. If a plant is designed to burn a low-grade fuel, and it is desired to change to a fuel of better grade, this can be done by reducing the size of the grate by bricking off a portion of it. With a better fuel less coal would be burned, and it would probably be burned at a higher rate; consequently so large a grate would not be needed. It might seem that it would not be necessary to reduce the size of the grate to burn less fuel, but, although less fuel could be burned per square foot of grate by reducing the draft, yet it would probably not be good practice to do so, as it is necessary to burn some coals at a high rate of combustion to secure the best result. Too slow combustion results in the partial burning of the gases, and this causes a loss. If it is certain, before a plant is constructed, that there never will be a desire to operate it with a lower grade of fuel than that for which it is designed, it would be foolish to provide larger grates than are necessary for this fuel, as boilers so constructed are frequently more expensive both in cost of construction and land occupied. A high rate of combustion is undoubtedly the best for many coals, for the reason that the gases are much more thoroughly consumed when the furnace temperature is high.

**Proportioning Grate.** — The ratio of the grate to heating surface varies with the kind of coal and the amount burned, as was explained in an earlier paragraph. From a knowledge of the number of pounds of water to be evaporated, and the amount one pound of a given kind of coal will evaporate under ordinary conditions, the quantity of coal which must be burned in a given time can be calculated. If one knows the number of pounds of coal that must be burned, and the amount of coal that should be burned on each square foot of grate surface, in a given time for the normal rate of evaporation, the amount of grate surface can be determined.

It remains, then, to provide sufficient draft, either by a fan or chimney, to produce not only the proper rate of combustion for ordinary demands, but also a higher one which will enable the boiler to operate with such overload as may be thought necessary. The relative evaporative power of the better grades of a number of different coals is shown in the table, Pocahontas coal being placed at 100. These figures are approximate and should be used with some caution. The relative evaporation for the different

coals shows what might be expected from the better grades of each kind of coal mentioned when fired by a good fireman under ordinary everyday conditions. The variation from these figures which might result from a difference in the quality of the coal from the same mine would be greatest with the Western coals, less with the Pennsylvania bituminous, and least with the semi-bituminous group. The anthracite, particularly the small sizes, might vary considerably on account of an abnormal amount of impurities present in the coal. The figures given in the table can be exceeded when all the conditions are favorable.

TABLE 3. — RELATIVE VALUE OF STEAM COALS.

| Kind of coal. | Relative evaporative power. | Pounds of water that 1 pound of coal will evaporate into steam from and at 212° F. under ordinary conditions. | Pounds of coal per square foot of grate per hour. | Draft between furnace and ash pit in inches of water. | Ratio of heating to grate surface. |
|---|---|---|---|---|---|
| Pocahontas, W. Va.* | 100 | 9.5 | 15 | .3 | 45 |
| Youghiogheny, Pa.† | 91.6 | 8.7 | 17 | .3 | 48 |
| Hocking Valley, O.† | 80 | 7.6 | 18 | .3 | 45 |
| Big Muddy, Ill.† | 80 | 7.6 | 20 | .3 | 50 |
| Mt. Olive, Ill.† | 67.5 | 6.4 | 20 | .3 | 45 |
| Lackawanna, Pa.,‡ pea | 84 | 8.0 | 15 | .5 | 35 |
| Lackawanna, Pa.,‡ No. 1 buckwheat | 79 | 7.5 | 13 | .6 | 32 |
| Lackawanna, Pa.,‡ rice | 74 | 7.0 | 12 | 1.0 | 30 |

* Semibituminous. † Bituminous. ‡ Anthracite.

Table 3 also shows about the amount of coal which should be burned per square foot of grate per hour under ordinary conditions; also, the ratio of heating to grate surface necessary for the boiler to develop its rated capacity when burning about the amount of coal stated and when each pound of coal is evaporating the quantities of water given in the table. Some authorities of considerable experience with Illinois coals advise higher rates of combustion than are recommended in the table for the best results. A grate proportioned in accordance with the data given can easily be reduced in size if found desirable. It is proposed to provide sufficient draft between the furnace and ash pit, that shown in the fourth column in the table, to run easily a boiler propor-

tioned according to the data given in the table at one-third over its rating. For the buckwheat and smaller sizes of anthracite mechanical draft should be used.

It is undoubtedly true that a better result will be obtained by higher rates of combustion than those given in the table, but if these are increased for the normal working of a boiler it will be necessary to have available a very intense draft, nothing less perhaps than that created by a high chimney or a fan, in order that there may be sufficient reserve draft to operate the boiler at much of an overload. In other words, if the rates of combustion are taken higher than those given for ordinary service, the draft must be made correspondingly greater to provide a reserve capacity in evaporative power. The high rates of combustion possible with mechanical draft are conducive to high furnace efficiency, but that is a subject which will be discussed later.

**Coal.** — In an earlier paragraph attention was called to the necessity of proportioning boilers to burn the fuel which will be the cheapest. It seems well, therefore, to show in a general way some of the properties and the relative values of different steam coals. Such information can be misleading on account of the variations that exist not only in different coals, but in coal mined from the same seam. Nevertheless an approximate relation can be given for the better grades of several different coals that are typical of the kinds most used. What the poorer grades of coal of some of these kinds will do, it is impossible to predict. The steam coals used in the eastern and middle parts of the United States may be divided into anthracite, bituminous, and semibituminous classes. A coal is classified in these groups according to the relative proportions of fixed carbon and volatile hydrocarbons that it contains. The hydrocarbons are those gases given off by certain coals when they are heated moderately. Semibituminous coals contain less than 25 per cent hydrocarbons and bituminous coals 25 to 60 per cent. The former are the best steam coals for the reason that when the hydrocarbons are more than 20 per cent of the fuel composition, the heat value of the fuel becomes less, for then the hydrocarbons contain more or less oxygen; while with less than 20 per cent hydrocarbon the volatile gases are mostly hydrogen, and the coal therefore has a higher heat value. The percentage of hydrocarbons in anthracite coal is very small.

**Semibituminous Coals.** — This group contains the finest steam coals mined in the United States. They are found mainly in Virginia, West Virginia, and Maryland. The ash varies from 3 to 8 per cent, while the coals contain about 14,500 heat units per pound. The coals of this group are much more uniform in heating power and evaporation than those of any other, but there is a variation in some of them, owing to the fact that care is not taken to exclude impurities which affect their heat value. This group of coals includes the Pocahontas, New River, Cumberland (George's Creek), and Clearfield varieties. The value is about in the order named, Pocahontas and New River probably being the most constant in quality. Placing the evaporative power of Pocahontas and New River at 100, none of the other coals would be hardly less than 95.

**Bituminous Coals.** — These are found in Pennsylvania, Ohio, Kentucky, Tennessee, Indiana, Illinois, Missouri, and other states. They differ widely in heating power, not only one coal from another, but a great difference is also found in coal from the same mine. The bituminous coals are divided into two classes, caking and noncaking. The Indiana, Illinois, and Missouri coals are of the caking variety, which on burning becomes pasty and forms into lumps that greatly impede the fire unless broken up. Certain Western coals have to be burned at a comparatively high rate of combustion, about 20 pounds of coal per square foot of grate per hour, otherwise it is difficult to keep the fire from going out. The Pennsylvania coals are much the best for steam purposes, and the Ohio coals are usually better than those found farther west.

**Anthracite Coal.** — This is mined chiefly in Pennsylvania, although quite a little is found in the Far West. It is principally composed of pure carbon, and its heat value is dependent mainly on the amount of earthy matter mixed with it when sold. The percentage of earthy matter is naturally greater in the smaller sizes. Anthracite coal is classified according to size into lump, broken, egg, stove, chestnut, pea, numbers 1 and 2 buckwheat, rice, and barley. Pea coal is about as large as is usually used for steam purposes. The larger sizes of this coal, that is, the chestnut size and over, are about equivalent in evaporating power to Pittsburgh bituminous coal. The smaller sizes require a very strong draft because the particles of coal, being small, pack to-

gether so that the air cannot get through the bed of fuel to cause rapid combustion. It is therefore impossible with natural draft to burn more than a very limited amount per square foot of grate, and it is inconvenient and costly to provide boilers with a sufficient grate to burn buckwheat or the smaller sizes with the draft due to an ordinary chimney. It is necessary, therefore, if this grade of fuel is to be used, to construct a 150-foot, preferably 200-foot chimney, or to employ mechanical draft. Rice and the smaller sizes cannot be burned without mechanical draft.

# CHAPTER III.

## DESIGN OF HORIZONTAL RETURN TUBULAR BOILERS AND BOILER SPECIFICATIONS.

The horizontal return tubular boiler is used to a far greater extent than any other type in the United States at the present time, and it is intended in this chapter to give some general rules for designing boilers of this type. Another well-known boiler of the fire-tube type is the Manning, which is a vertical boiler with unusually long tubes rising from a high combustion chamber surrounded by water. It has been used successfully to a considerable extent in New England. Various boilers of the fire-tube type of special design have been illustrated from time to time in technical papers, but the use of these boilers is so limited compared to the horizontal return tubular boilers that they will not be further alluded to. A sectional view of a typical setting for a horizontal return tubular boiler is given in Fig. 5. The method of proportioning the different parts of a boiler of this type follows:

**Thickness of Shell.** — One of the most satisfactory rules for determining the thickness of shell necessary in boilers of the fire-tube type is that used by the steam-boiler inspection department of the City of Philadelphia. The rule is as follows:

$$\text{Working pressure} = 2\,Tts \div Df,$$

in which

$T$ = the ultimate tensile strength of the plate in pounds per square inch;
$t$ = the thickness of the plate in inches;
$s$ = the efficiency of the longitudinal joint;
$D$ = the diameter of the boiler in inches;
$f$ = the factor of safety.

The factor of safety is usually taken at 5. With a tensile strength of 60,000 pounds and a joint efficiency of 80 per cent, the rule shows that with a $\frac{9}{16}$-inch plate, which is about as thick

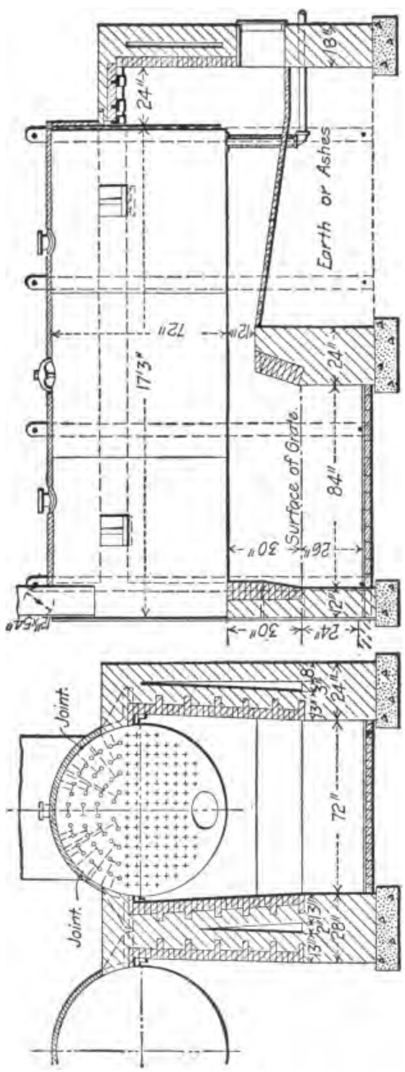



Fig. 5. Typical Setting of Horizontal Return Tubular Boiler.

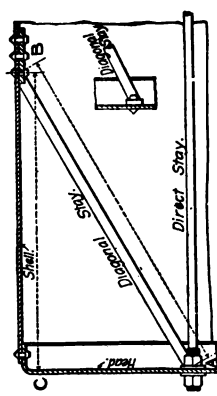



Fig. 7. Diagonal and Direct Stays.

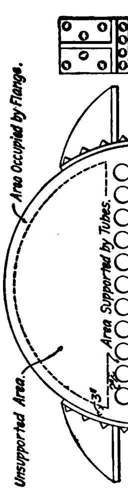



Fig. 6. Unsupported Part of Boiler Head.

as is commonly used for the shell of an externally fired boiler on account of the resistance that a thicker plate offers to the transfer of heat at the girth seams of the boiler, the highest working pressure that can be carried in a boiler 6 feet in diameter is 150 pounds. A few horizontal return tubular boiler plants have been designed to carry a steam pressure of 150 pounds, but it is better to use water-tube or internally fired fire-tube boilers if this or a higher pressure is to be carried unless the boilers are designed by an expert.

**Braces.** — Braces are used in horizontal tubular boilers to balance the pressure exerted on those parts of the heads of the boiler above and below the tubes and not close to the shell. If these parts are not braced, the pressure would cause the flat head to bulge outward. For a distance of 3 inches from the shell, and for a distance of 2 inches from the tubes, the head is usually considered to be sufficiently stiffened by the circumferential joint and the tubes not to need other bracing. Fig. 6 shows one of the heads in a horizontal tubular boiler, and the part which has to be braced is indicated. Braces used in this work are usually of two kinds, —the direct or through stay or brace, which extends entirely

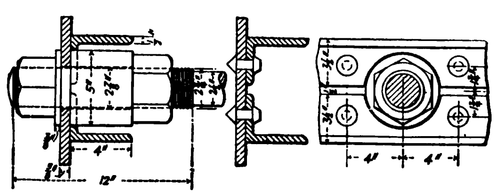


Fig. 8. Connecting Direct Braces.

through the boiler and joins the heads together, and the diagonal or crowfoot brace. This latter is shown in Fig. 7. The head is connected to the adjacent shell by it. The diagonal brace is preferred by the boiler-inspection companies, as it is much easier to get inside of a boiler to inspect the interior with this brace than with the direct type. If the latter are used they should be sufficiently separated to allow a man entering the manhole opening to pass between them. The braces should be distributed uniformly over the area they are intended to support. Fig. 5 shows the manner in which diagonal braces may be distributed over the head. Five direct braces are sometimes used, three being placed in a lower row and two above, all being symmetrically placed with

reference to the center. When direct braces are used the heads are frequently further stiffened by channel iron or angle irons riveted to the heads as shown in Fig. 8. If manholes are placed in one of the heads of the boiler, the heads below the tubes should be braced.

In determining the number and size of braces, the area in square inches of the surface to be stayed must be calculated and then the total pressure on this area. This latter divided by 7500 will give the cross-sectional area in square inches that must be provided in all of the braces combined, 7500 pounds per square inch being the greatest strain to which a brace should be subjected. Cutting a thread on the end of the braces for the nuts reduces the area of metal somewhat, and this should be taken into account. Sometimes the ends of the braces are upset, that is, heated and hammered at the end so that their diameter is increased slightly where the thread is to be cut.

With diagonal braces the pull exerted is not perpendicular to the head of the boiler; hence the area of the brace should be

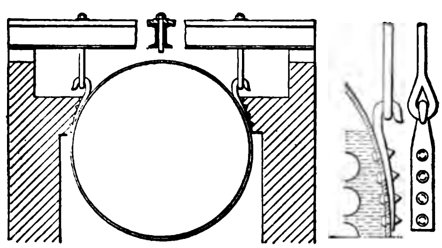


Fig. 9. Method of Suspending Boiler.

calculated by dividing the surface supported by 7500 and then increasing the result by multiplying the quotient obtained by the length in inches of the brace divided by the distance in inches that the head is from the point where the brace is attached to the shell. Referring to Fig. 7, the length of the brace is the distance A B, and the distance from the head to the point where the brace is attached is the distance C B. The ratio A B : C B is the amount that the brace should be increased. For instance, if the surface to be stayed should require the pull of a brace 1 square inch and the length A B should be 48 inches and the length C B 40 inches, then the area of brace actually required would be $1 \times 48 \div 40 = 6 \div 5 = 1\frac{1}{5}$ square inches.

**Supporting Boilers.**— For many years the practice has been to support boilers by riveting two pairs of lugs on the sides of the shell, as in Fig. 6, and supporting them on cast-iron plates built into the setting, rollers being placed under the pair of lugs in the rear, so that the movement due to expansion will be toward the rear. The principal objection to this plan is that the settlement of the walls of the setting, which is almost sure to occur, often throws almost the entire work of supporting the boiler on two lugs. A number of engineers suspend a boiler by riveting straps to the shell and passing hooked rods through them and through cast-iron plates resting on I beams laid across the top of the setting. Fig. 9 shows this construction. This is one of the best methods. If it is used it is well to have an iron plate of generous size between the beams and the setting for the former to rest upon.

TABLE 4. — TUBE SPACING AND HEATING SURFACE IN HORIZONTAL TUBULAR BOILERS.

| Diameter boiler, inches. | Tubes. | | | Heating surface per linear foot of boiler, square feet. | Size of manholes, inches. | Tubes, center to center. | | Center of boiler to center of upper row tubes, inches. | Usual length of boiler, feet. |
|---|---|---|---|---|---|---|---|---|---|
| | Diameter, inches. | Number. | Internal area, square feet. | | | Horizontal, inches. | Vertical, inches. | | |
| A. | B. | | | | D. | E. | F. | G. | |
| 48 | 3 | 46 | 1.94 | 43.67 | 9×14½ | 4¼ | 4 | 6½ | 10 to 12 |
| 50 | 3 | 52 | 2.19 | 48.69 | 9×14½ | 4¼ | 4 | 6¾ | 10 to 12 |
| 52 | 3 | 54 | 2.28 | 50.58 | 9×14½ | 4¼ | 4 | 7¼ | 10 to 12 |
| 54 | 3 | 60 | 2.53 | 55.60 | 9×14½ | 4½ | 4 | 7½ | 10 to 12 |
| 56 | 3 | 64 | 2.70 | 59.06 | 11×15 | 4½ | 4 | 8 | 10 to 12 |
| 58 | 3 | 70 | 2.95 | 64.09 | 11×15 | 4½ | 4 | 8¼ | 10 to 12 |
| 60 | 3 | 76 | 3.21 | 69.11 | 11×15 | 4½ | 4 | 8½ | 10 to 12 |
| 48 | 3½ | 34 | 1.97 | 38.68 | 9×14½ | 5 | 4½ | 6¼ | 12 to 14 |
| 50 | 3½ | 38 | 2.20 | 42.66 | 9×14½ | 5 | 4½ | 6½ | 12 to 14 |
| 52 | 3½ | 46 | 2.67 | 50.31 | 9×14½ | 5 | 4½ | 7¼ | 12 to 14 |
| 54 | 3½ | 47 | 2.73 | 51.53 | 9×14½ | 5 | 4½ | 7¼ | 12 to 14 |
| 56 | 3½ | 50 | 2.90 | 54.60 | 11×15 | 5 | 4½ | 7½ | 12 to 14 |
| 58 | 3½ | 52 | 3.02 | 56.74 | 11×15 | 5¼ | 4½ | 7¾ | 12 to 14 |
| 60 | 3½ | 56 | 3.25 | 60.71 | 11×15 | 5¼ | 4½ | 8¼ | 14 to 16 |
| 62 | 3½ | 60 | 3.48 | 64.72 | 11×15 | 5¼ | 4½ | 8½ | 14 to 16 |
| 64 | 3½ | 64 | 3.71 | 68.67 | 11×15 | 5½ | 4½ | 9 | 14 to 16 |
| 66 | 3½ | 70 | 4.06 | 74.49 | 11×15 | 5½ | 4½ | 9¼ | 14 to 16 |
| 54 | 4 | 36 | 2.75 | 46.17 | 11×15 | 6 | 5 | 7 | 16 to 20 |
| 56 | 4 | 38 | 2.90 | 48.59 | 11×15 | 6 | 5 | 7¼ | 16 to 20 |
| 58 | 4 | 40 | 3.05 | 50.99 | 11×15 | 6 | 5 | 7¾ | 16 to 20 |
| 60 | 4 | 47 | 3.59 | 58.63 | 11×15 | 6 | 5 | 8 | 16 to 20 |
| 62 | 4 | 49 | 3.74 | 61.04 | 11×15 | 6 | 5 | 8¼ | 16 to 20 |
| 64 | 4 | 51 | 3.89 | 63.45 | 11×15 | 6 | 5 | 8¾ | 16 to 20 |
| 66 | 4 | 56 | 4.27 | 69.00 | 11×15 | 6 | 5 | 9 | 16 to 20 |
| 68 | 4 | 62 | 4.73 | 75.59 | 11×15 | 6 | 5 | 9½ | 16 to 20 |
| 70 | 4 | 64 | 4.88 | 78.01 | 11×15 | 6 | 5 | 9¾ | 16 to 20 |
| 72 | 4 | 74 | 5.65 | 88.79 | 11×15 | 6 | 5 | 10¼ | 16 to 20 |

**Tubes.**—Tubes should not be so closely spaced in a boiler as to interfere with a proper circulation of water in them. Three $3\frac{1}{2}$- and 4-inch tubes are commonly used in horizontal tubular boilers, the last being generally used in large-size boilers, the size decreasing with the diameter of the boiler. Barrus states that a certain ratio of tube area to grate surface is necessary to give the best result with different fuels, a ratio of nine to one or ten to one being proper for anthracite coals and six to one or seven to one for bituminous coals, the difference being due to the large volume of gases with bituminous coals, which requires a large area in the tubes to pass the gases at the proper velocity for giving off their heat. Tubes should be placed in vertical rows and not staggered. They should be located as far apart as the number necessary to put in the boiler will permit; for this will permit a better circulation, which is essential when the boiler is operating at high rates of combustion.

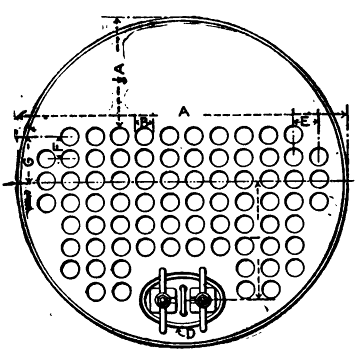



Fig. 10.

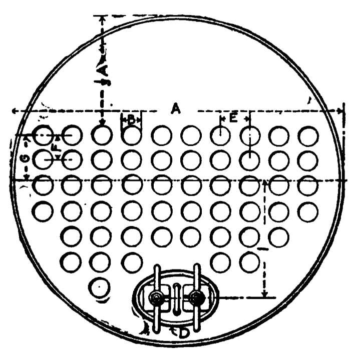



Fig. 11.

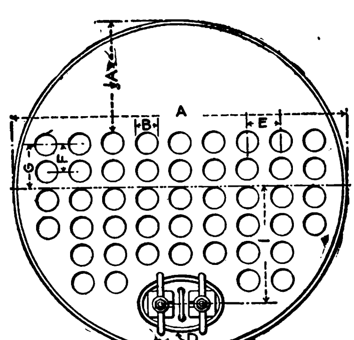



Fig. 12.

By means of Table 4 and Figs. 10, 11, and 12 the proper method of locating and spacing tubes in return tubular boilers may be found. The three figures refer respectively to 60-inch boilers with 3-, 3½- and 4-inch tubes. These data have been taken from Mr. W. M. Barr's work, "Boilers and Furnaces." Fig. 8 has also been taken from the same source. In using Table 4 to determine the number and location of tubes for boilers of different diameters than those shown in the figure, a drawing of the tube sheet should be made on which the centers of the tubes can be located from the data given in the tables. Barr states that "the distance from the side of a tube to the inside of the boiler shell should not, in the case of a 36-inch boiler, be less than 2 inches, for a 48-inch boiler it should not be less than 2½ inches, and for

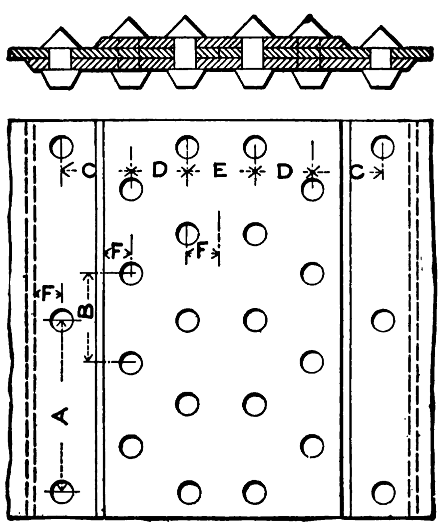



Fig. 13.

a 50-inch boiler and larger, the distance should be not less than 3 inches, to secure good water circulation." Calculations have been made showing the heating surface per lineal foot of boiler for each of the different sizes given in the table, also the usual length of each. The tube area is also given, and this may be useful in comparing the tube area with the size of grate. The method of determining the proper amount of heating and grate surface was explained in the preceding chapter.

## TABLE 5. — DIMENSIONS OF DOUBLE-RIVETED STAGGERED SEAMS.

| Thickness of sheet, inches. | Diameter of rivets, inches. | Diameter of rivet holes, inches. | Pitch, inches. | Lap, inches. | Distance between rows of rivets, inches. | Edge of sheet to pitch line, inches. | Efficiency. — Weakest part. |
|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $\frac{11}{16}$ | $\frac{3}{4}$ | $2\frac{7}{8}$ | $4\frac{3}{16}$ | $1\frac{15}{16}$ | $1\frac{1}{8}$ | 0.739 |
| $\frac{5}{16}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $2\frac{7}{8}$ | $4\frac{3}{8}$ | $1\frac{15}{16}$ | $1\frac{7}{32}$ | 0.717 |
| $\frac{3}{8}$ | $\frac{7}{8}$ | $\frac{15}{16}$ | $3\frac{1}{4}$ | 5 | $2\frac{3}{16}$ | $1\frac{13}{32}$ | 0.711 |
| $\frac{7}{16}$ | $\frac{15}{16}$ | 1 | $3\frac{5}{16}$ | $5\frac{3}{16}$ | $2\frac{3}{16}$ | $1\frac{1}{2}$ | 0.687 |
| $\frac{1}{2}$ | 1 | $1\frac{1}{16}$ | 3.32 | $5\frac{1}{2}$ | $2\frac{5}{16}$ | $1\frac{19}{32}$ | 0.677 |
| *Single-riveted Girth Seams Used with Above Longitudinal Seams.* | | | | | | | |
| $\frac{1}{4}$ | $\frac{11}{16}$ | $\frac{3}{4}$ | $2\frac{1}{16}$ | $2\frac{1}{4}$ | ......... | ......... | 0.545 |
| $\frac{5}{16}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $2\frac{1}{8}$ | $2\frac{7}{16}$ | ......... | ......... | 0.494 |
| $\frac{3}{8}$ | $\frac{7}{8}$ | $\frac{15}{16}$ | $2\frac{3}{8}$ | $2\frac{13}{16}$ | ......... | ......... | 0.49 |
| $\frac{7}{16}$ | $\frac{15}{16}$ | 1 | $2\frac{7}{16}$ | 3 | ......... | ......... | 0.466 |
| $\frac{1}{2}$ | 1 | $1\frac{1}{16}$ | $2\frac{1}{2}$ | $3\frac{3}{16}$ | ......... | ......... | 0.449 |

## TABLE 6. — TRIPLE-RIVETED LAP JOINT.

| Thickness of sheet, inches. | Diameter of rivets, inches. | Diameter of rivet holes, inches. | Pitch, inches. | Lap, inches. | Distance between rows of rivets, inches. | Edge of sheet to pitch line, inches. | Efficiency. —Weakest part. |
|---|---|---|---|---|---|---|---|
| $\frac{1}{4}$ | $\frac{5}{8}$ | $\frac{11}{16}$ | 3 | $6\frac{1}{16}$ | 2 | $1\frac{1}{32}$ | 0.77 |
| $\frac{5}{16}$ | $\frac{11}{16}$ | $\frac{3}{4}$ | $3\frac{1}{8}$ | $6\frac{3}{8}$ | $2\frac{1}{16}$ | $1\frac{1}{8}$ | 0.76 |
| $\frac{3}{8}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $3\frac{1}{4}$ | $6\frac{13}{16}$ | $2\frac{3}{16}$ | $1\frac{7}{32}$ | 0.75 |
| $\frac{7}{16}$ | $\frac{7}{8}$ | $\frac{15}{16}$ | $3\frac{3}{4}$ | $7\frac{13}{16}$ | $2\frac{1}{2}$ | $1\frac{13}{32}$ | 0.75 |
| $\frac{1}{2}$ | $\frac{15}{16}$ | 1 | $3\frac{15}{16}$ | $8\frac{1}{4}$ | $2\frac{5}{8}$ | $1\frac{1}{2}$ | 0.746 |
| *Single-riveted Girth Seams Used with Above Longitudinal Seams.* | | | | | | | |
| $\frac{1}{4}$ | $\frac{5}{8}$ | $\frac{11}{16}$ | $2\frac{1}{16}$ | $2\frac{1}{16}$ | ......... | ......... | 0.456 |
| $\frac{5}{16}$ | $\frac{11}{16}$ | $\frac{3}{4}$ | $2\frac{1}{8}$ | $2\frac{1}{4}$ | ......... | ......... | 0.419 |
| $\frac{3}{8}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $2\frac{1}{8}$ | $2\frac{7}{16}$ | ......... | ......... | 0.412 |
| $\frac{7}{16}$ | $\frac{7}{8}$ | $\frac{15}{16}$ | $2\frac{3}{8}$ | $2\frac{13}{16}$ | ......... | ......... | 0.42 |
| $\frac{1}{2}$ | $\frac{15}{16}$ | 1 | $2\frac{1}{2}$ | 3 | ......... | ......... | 0.398 |

TABLE 7. — TRIPLE-RIVETED BUTT JOINT WITH DOUBLE WELT.

| Thickness of sheet, inches. | Diameter of rivets, inches. | Diameter of rivet holes, inches. | Thickness of strap, inches. | A. | B. | C. | D. | E. | F. | Efficiency. —Weakest part. |
|---|---|---|---|---|---|---|---|---|---|---|
| $\frac{5}{16}$ | $\frac{11}{16}$ | $\frac{3}{4}$ | $\frac{1}{4}$ | $6\frac{1}{4}$ | $3\frac{1}{8}$ | $2\frac{3}{8}$ | $2\frac{1}{8}$ | $2\frac{1}{2}$ | $1\frac{1}{4}$ | 0.88 |
| $\frac{3}{8}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | $\frac{5}{16}$ | $6\frac{1}{2}$ | $3\frac{1}{4}$ | $2\frac{1}{2}$ | $2\frac{3}{16}$ | $2\frac{7}{16}$ | $1\frac{7}{32}$ | 0.75 |
| $\frac{7}{16}$ | $\frac{7}{8}$ | $\frac{15}{16}$ | $\frac{3}{8}$ | $6\frac{3}{4}$ | $3\frac{3}{8}$ | $2\frac{3}{4}$ | $2\frac{1}{4}$ | $2\frac{13}{16}$ | $1\frac{13}{32}$ | 0.86 |
| $\frac{1}{2}$ | $\frac{15}{16}$ | 1 | $\frac{7}{16}$ | $7\frac{1}{2}$ | $3\frac{3}{4}$ | 3 | $2\frac{3}{8}$ | 3 | $1\frac{1}{2}$ | 0.866 |

*Single-riveted Girth Seams Used with Above Longitudinal Seams.*

| Thickness of sheet, inches. | Diameter of rivets, inches. | Diameter of rivet holes, inches. | Pitch, inches. | Lap, inches. | Efficiency. —Weakest part. |
|---|---|---|---|---|---|
| $\frac{5}{16}$ | $\frac{11}{16}$ | $\frac{3}{4}$ | 2 | $2\frac{1}{4}$ | 0.446 |
| $\frac{3}{8}$ | $\frac{3}{4}$ | $\frac{13}{16}$ | 2 | $2\frac{7}{16}$ | 0.438 |
| $\frac{7}{16}$ | $\frac{7}{8}$ | $\frac{15}{16}$ | $2\frac{1}{4}$ | $2\frac{13}{16}$ | 0.444 |
| $\frac{1}{2}$ | $\frac{15}{16}$ | 1 | $2\frac{1}{4}$ | 3 | 0.442 |

**Riveting.** — In Tables 5, 6, and 7 there are given proportions of riveted joints recommended by the Hartford Steam Boiler Inspection and Insurance Company, which are published through the courtesy of that corporation. The efficiencies for the different joints, given in the table, are to be substituted in the formula for determining the thickness of shell plates. The dimensions to which the letters in Table 7 refer are shown in Fig. 13. Butt joints are far superior to lap joints, as there is not the tendency of the plate to crack on a line parallel with and near the rivets. The joints are designed for metal having a tensile strength of from 55,000 to 60,000 pounds, and the efficiency calculations are based upon the latter figure. The shearing strength allowed to the rivet steel per square inch of section in single shear, when used in steel plates, is 38,000 pounds, and a rivet in double shear is considered to be equivalent to 85 per cent additional. Rivet steel is allowed for shearing strength 45,000 pounds per square inch of section. In the calculations for the efficiency the rivet is assumed to fill the hole, and the diameter of the rivet hole, not that of the rivet, is therefore used. The efficiency of the girth seams given in the tables need not be considered in determining the thickness of shells of boilers.

**Boiler Settings.** — Fig. 5 shows a longitudinal and cross section of popular form of setting for return tubular boilers. It should be constructed of hard red brick, laid in lime or cement mortar, and the entire setting ought to be lined with fire brick, with every fifth course laid as headers, so that any part that might become damaged could be easily renewed without taking out the entire lining. Sometimes, to economize, the furnace as far as the bridge wall only is lined with fire brick. The fire brick should be laid in fire clay, using as little of it as possible. In the drawing the grate has a width equal to the diameter of the boilers, and the side wall is battered so as to leave a space of 3 inches at the level where the setting closes into the boilers. The side walls are provided with air spaces, as shown, which are necessary to prevent the wall from cracking.

If fire brick backed with common brick is used, the side walls have to be 13 inches thick, and the air space should be about 2 inches wide at the bottom and diminish as its sides converge. Two 13-inch walls and a 2-inch air space make the entire thickness of the wall between the boilers 28 inches. This diminishes at the top as shown. It is sometimes difficult with low-grade fuels and natural draft to burn sufficient coal on the grate of a horizontal tubular boiler to obtain as high evaporation as might be needed, hence it is desirable, if these boilers are apt to be fired with such fuel, to place the boilers a little farther apart and make the side walls perpendicular and at a distance apart at the grate equal to the diameter of the boiler plus 6 inches. This makes the grate 6 inches wider and increases its surface materially. If a boiler is constructed in this way, it is a simple matter to diminish the size of the grate by bricking off the sides and rear. It is difficult to fire a grate more than 7 feet deep, although they are sometimes made 9 feet in depth, in large boilers. Furnaces over 5 feet wide should have two fire- and two ash-pit doors.

The top of the bridge wall of the boiler is usually about 10 or 12 inches from the bottom of the shell, and the space behind may be filled with earth and paved with common brick or left empty. Curving this combustion chamber to conform with the shell only reduces its size, which is a disadvantage with bituminous coal and of no use with other kinds. The rear wall should contain an air space and be provided with a clean-out door about 16 by 20 inches. The wall should be located at a distance of 18

to 24 inches, depending on the size of the boiler, from the back head.

The back connection, that is, the connection between the rear wall and the head, is a source of more or less trouble on account of the expansion and contraction of the boiler, and the difficulty of making a joint that will remain tight. One method is to spring an arch across, having one end resting on the wall and the other upon an angle iron riveted to the back head of the boiler. The arch consists of brick resting in an iron framework. The Hartford Steam Boiler Inspection and Insurance Company uses an arch somewhat similar to that shown in Fig. 5.

The grate should be at a level of 24 inches above the floor and the shell from 28 to 30 inches above the grate with anthracite coals and from 36 to 42 inches if bituminous coals are to be burned, as the gases from the latter do not burn properly if the flame comes in contact with the cooler boiler surfaces. The floor of the ash pit should be paved with brick laid on edge and flushed with a thin cement mortar so as to be water-tight. This floor is usually 3 to 6 inches below the floor of the boiler room.

An interesting horizontal tubular boiler is shown in Plate 4 and Fig. 14. This boiler was built for a steam pressure of 185 pounds per square inch from the plans and specifications of Dr. E. D. Leavitt, consulting engineer. A full description of it appeared in the *Engineering Record* of May 18, 1901.

**Boiler Specifications.** — It was thought advisable to print one or two typical specifications for horizontal return tubular boilers and to outline a method to be pursued in purchasing water-tube boilers, the difference in construction of boilers of the latter type making it necessary for the specification to be mainly a general description of what is wanted and what the boilers are to do.

In the first form of specification printed, which calls for the delivery and setting of a boiler, those clauses that relate to the prosecution of the work, the removal of rubbage after its completion, remuneration for a change in the plans, interpretation of drawings, etc., are omitted. The specification is divided into sections, and in some cases comment or explanation follows a section. A form of specification for high-grade work follows:

*Intent of Specifications.* — It is the intent and purpose of these specifications to provide for the furnishing and installation of a complete boiler plant comprising the boilers, setting, grates, fronts, feed and blow-off piping, gauges,

smoke breeching, flues, dampers, and such details as are hereinafter specified, all complete and erected ready to generate steam.

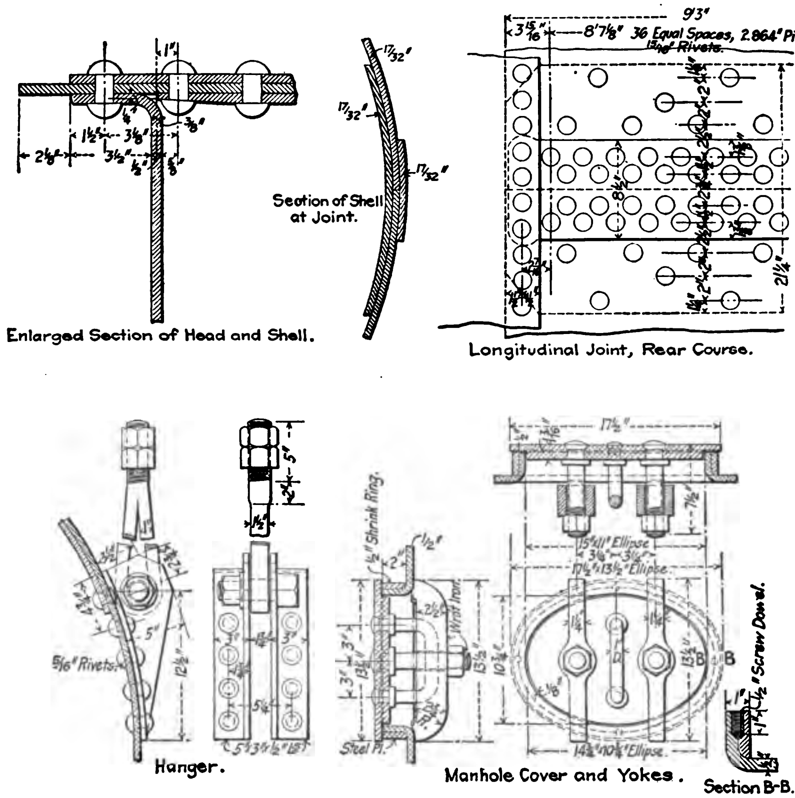



Fig. 14. Details of Leavitt Boiler.

*Materials.* — The shells and heads of the boiler are to be constructed of open-hearth steel having a tensile strength of from 55,000 to 65,000 pounds per square inch, with a yield point not less than one-half the tensile strength and an elongation of 25 per cent in 8 inches. (See note below.) Test specimens cut from the plates must, when cold, be capable of being bent 180 degrees flat on themselves without fracture on the outer side of the bent portion, also, after being heated to a light cherry red and quenched in water of a temperature between 80° and 90° F. For the bending tests the test specimens shall be $1\frac{1}{2}$ inches wide, if possible, and for all material $\frac{3}{4}$ inch or less in thickness, the test specimen shall be of the same thickness as that of the finished material from which it is cut; but for material more than $\frac{3}{4}$ inch thick, the bending specimen may be only $\frac{1}{2}$ inch thick. One specimen for the cold bending and one for the quenched test to be furnished from each plate and

marked for identification. Specimens for the tensile tests of the dimensions shown in the upper illustration in Fig. 15 are to be cut from each plate, marked for identification, and given to the engineer.

Rivet steel shall have a tensile strength of from 45,000 to 55,000 pounds per square inch, the yield point to be not less than one-half the tensile strength and the elongation 28 per cent in 8 inches. Rivet steel of the full size as rolled shall pass the same bending tests as specified for shell plates. Two tensile test specimens shall be furnished from each melt of rivet rounds. In

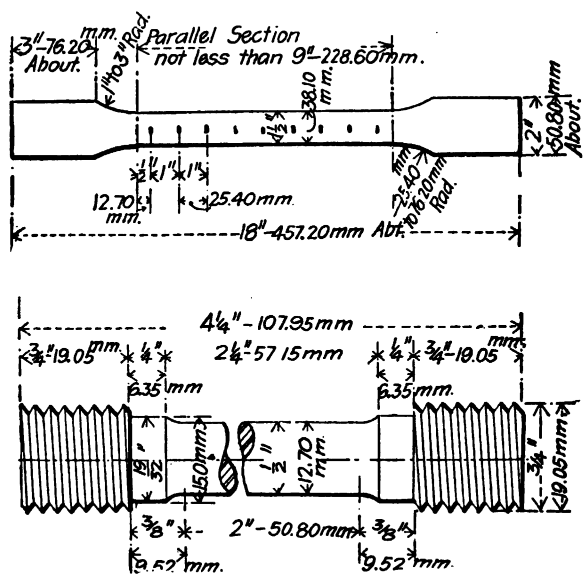



Fig. 15. Dimensions of Test Specimens.

case any one of these develops flaws or breaks outside of the middle third of its gauged length, it may be discarded and another test specimen substituted therefor. Two cold bending specimens and two quenched bending specimens shall be furnished from each melt of rivet rounds.

Note. — The foregoing section upon materials is an abstract of the standard specification for boiler plate and rivet steel adopted June 29, 1901, by the American Section of the International Association for Testing Materials. Reference to the chemical properties of the steel has been omitted. A variation in the elongation with different thicknesses of plate is provided as follows: For each increase of $\frac{1}{8}$ inch in thickness above $\frac{3}{4}$ inch, a deduction of one per cent shall be made from the specified elongation. For each decrease of $\frac{1}{16}$ inch in thickness below $\frac{5}{16}$ inch a deduction of $2\frac{1}{2}$ per cent shall be made in the specified elongation.

*Shell and Heads.* — The shell plates are to be — inches thick, the heads — inches thick. The shell is to be in three courses (or two courses if the boiler is small enough), and the middle one must be of the smallest diameter. The

edges of the plates must be planed before they are put together and calking must be done with a round-nosed tool. Heads must be flanged to a radius not less than $1\frac{1}{2}$ inches and they must be annealed after flanging. The flange must be free from cracks and flaws.

*Riveting.* — The girth seams are to be of the single-riveted lap-joint type, the pitch of the rivets to be — inches, lap — inches, and rivets — inches in diameter. The longitudinal seams are to be placed well above the fire line and to be of the butt type triple riveted with double welt, as shown in the drawing (which the engineer should supply). The rivet holes are to be drilled or punched $\frac{1}{8}$ inch small and afterward drilled or reamed out to full size. A reamer must be used instead of the drift-pin. The bur is to be removed and the rivets driven by a hydraulic or pneumatic riveter as far as possible.

NOTE. — Some specifications intended for high-grade work demand that the rivet holes shall be punched $\frac{1}{8}$ inch small and the shell then rolled and the lapping ends bolted together while the holes are drilled out the full size. This insures that the holes in one end of the sheet shall come opposite those in the other end, and much better riveting will be the result. Oftentimes the holes are punched out to the full size, but authorities unite in saying that the metal immediately surrounding a punched hole is weakened. Hence it is better to punch the holes small and drill or ream them out to full size.

*Bracing.* — The heads are to be braced to the shell by — radial solid crow-foot braces, — on each head distributed as shown on drawing. The braces must be of — inch stay-bolt iron at least 3 feet long, forged up without a weld and be connected to the shell by two — inch rivets. Each brace must be so located as to lie in a plane passing through the axis of the boiler.

NOTE. — If through braces are used, their number and diameter should be given and their location shown on a drawing. If the ends of the braces are to be upset, or if channel or angle irons are to be riveted to the heads to stiffen them, it should be so stated, and their weight and dimensions specified, or shown by a drawing.

*Tubes.* — The tubes are to be — in number, — inches in diameter, — feet long, of the best charcoal iron lap welded or of steel. The tube holes in the head are to be chamfered and the tubes are to be carefully expanded and beaded over at the ends. The tubes are to be located as shown in a drawing (which the engineer should supply).

*Manholes.* — A pressed steel manhole frame, 11 by 15 inches, is to be riveted to the top of each boiler, with the long diameter girthwise (in a position indicated by the engineer), and another is to be riveted to the front head below the tubes. The usual pressed steel manhole plate, gasket, yoke, and bolt are to be provided for each.

*Handhole.* — A 4-by-6-inch handhole is to be placed in the back head below the tubes. The plate, gasket, yoke, and bolt are to be provided.

*Nozzles.* — Each boiler is to have two — inch cast-iron nozzles riveted to the boiler in the middle of the front and rear courses. The nozzles are to be drilled to fit the A. S. M. E. standard flange schedule.

*Lugs.* — The boiler is to be supported on the brickwork by four heavy cast-iron lugs riveted to the shell, one pair to the front and the other to the

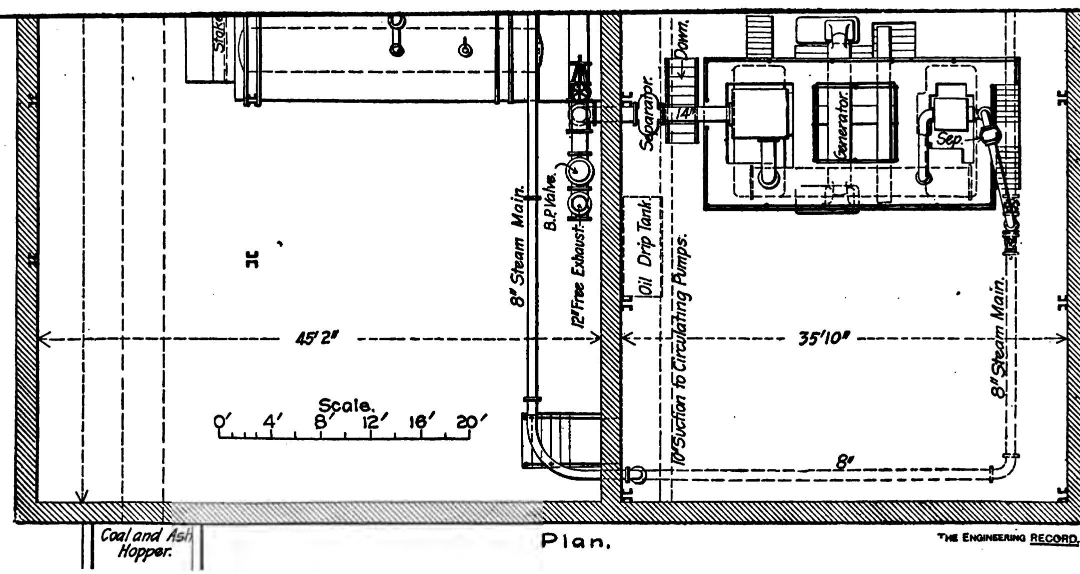



PLATE 5. — POWER STATION AT NAVY YARD, BROOKLYN, N. Y.

REAR-ADMIRAL GEORGE W. MELVILLE, CHIEF ENGINEER.

rear course. The front lugs shall rest upon bearing plates. Three 1-inch rollers shall be placed under each rear lug between it and the bearing plates, which are also to be provided.

NOTE. — If the boiler is to be supported by straps from beams laid across the top of the setting, it should be specified and a drawing showing the detail of the method of hanging the boiler given.

*Feed Pipe.* — The contractor is to supply and connect a — inch steel (or brass) feed pipe that is to pass through a brass bushing in the front head of each boiler, at one side and 3 inches above the tubes and extending to the rear of the boiler.

*Tools.* — Each boiler is to be provided with a fire shovel, slice bar, hoe, and poker. The boilers are to be provided with one tube scraper and the necessary hose and nozzle for blowing soot from the interior of the tubes.

*Blow-off.* — A blow-off connection is to be provided at the bottom of each shell at the back end. The — inch opening in the shell is to be tapped to receive a — inch blow-off pipe and the opening is to be reënforced by a — inch steel plate riveted to the shell. The blow-off pipe is to be protected by a fire-clay tube and run in the manner shown in the drawing to a blow-off main provided for in another contract. The boiler contractor is to supply a — inch blow-off cock of — make for each boiler and the fire-clay tube for protecting the blow-off pipe.

*Castings.* — Each boiler is to be provided with a cast-iron flush front, cheek plates, mouth plates, one (or two) fire and one (or two) ash-pit doors and clean-out door for rear wall, arch plates, grate bars, bearing bars, T bars for back connection, tie rods, wall braces, etc. The grate bars shall contain 50 per cent air space and are to fit a grate — by — inches and to be suitable for burning —— coal. The fire doors are to be fitted with registers and perforated linings, the ash-pit doors with registers, and the clean-out door with a lining.

*Fittings.* — Each boiler to be furnished with the following fittings, which are to be connected: One water column upon which can be mounted a steam gauge, gauge cocks, and gauge glass connected to the boiler by 1½-inch pipe; three — inch gauge cocks; one — inch gauge glass; one — inch steam gauge of — make and provided with stopcock and siphon.

*Hydrostatic Test.* — The boiler must be made sufficiently tight to stand without leaking when subjected to a hydrostatic test of a pressure 33⅓ per cent in excess of the normal working pressure of — pounds.

*Setting.* — The boilers are to be set in accordance with the accompanying drawing. The settings are to extend 6 inches below the floor level and are to be of the best brick laid in lime mortar and the entire setting is to be lined with the best quality of fire brick laid in fire clay and with every fifth course set as headers. The setting is to be closed in at the shell 5 inches above the center line of the boilers, except at the lugs where the setting closes in two courses below the lugs. The boilers are to be supported by lugs as previously stated, and care must be taken that bearing plates are placed level and that rollers under the rear and middle lugs are placed perpendicular to the axis of the boilers. The foundations for the setting are to be laid by the contractor.

**Smoke Flue and Chimney.** — If an iron chimney is to be used, some believe, in small work, that it is best to put the boilers, smoke breechings and flue, damper and damper regulator, and chimney in the boiler contract, for the reason that all of them affect the operation of the boiler. The method of proportioning the chimney, etc., is discussed later. The boiler specification, therefore, should indicate the thickness and dimensions of the smoke flue. Provisions should be made for a damper in the uptake of each boiler that may be closed when the boiler is undergoing repairs. There should be a damper in the main flue, and this should be under the control of an automatic damper regulator which opens or closes the damper according as the steam pressure is respectively below or above the normal. The smoke flue should have doors through which the soot may be removed when necessary.

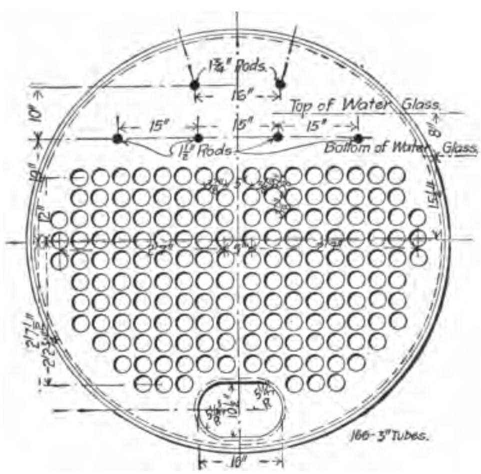



Fig. 16.

The following specification for a horizontal return tubular boiler, without the setting, was prepared by Messrs. Dean & Main, a well-known firm of mill engineers, and it is printed through their courtesy. The author believes it to be an excellent specification for the purpose intended. It will be noticed that it calls for the construction and delivery of the boilers only. Appended to the specification was a blue-print, reproduced in Fig. 16.

*Specifications for Three 78-inch Horizontal Return Tubular Boilers for ——*

*Proposals.* — Proposals will be received by Dean & Main, Exchange Building, Boston, Mass., for building and delivering three horizontal return tubular boilers of the following general dimensions:

| | |
|---|---|
| Working pressure by gauge, lbs. | 125 |
| Inside diameter of end rings of shell, in. | 78 |
| Thickness of shell, in. | $\frac{7}{16}$ |
| Length of tubes, ft. | 18 |
| Diameter of tubes, in. | 3 |
| Number of tubes | 166 |
| Width of grate, ft. | 7 |
| Length of grate, ft. | 7 |
| Height of center of boilers above floor, ft. | 8.25 |
| Distance from c. to c. of boilers, ft. | 9 |
| Height front of grate above floor, ft. | 2.5 |
| Height back of grate above floor, ft. | 2 |
| Thickness of heads, in. | $\frac{1}{2}$ |
| Diameter of rivets, in. | $\frac{7}{8}$ |
| Heating surface, sq. ft. | 2376 |
| Grate area, sq. ft. | 49 |

*Contract Price.* — The price is to be stated to cover the manufacture of the boilers and their delivery with the accessories furnished on cars.

*Quality of Materials and Workmanship.* — It is the intention that the materials and workmanship of the boilers shall be of the best. A representative of Dean & Main is to have the privilege of inspecting the boilers during their construction.

*Plates.* — The plates are to be of the best quality of open hearth acid or basic steel, having the following qualities:

| | |
|---|---|
| Elastic limit per sq. in., not less than | 30,000 lbs. |
| Ultimate strength per sq. in. | 52,000 to 60,000 lbs. |
| Elongation in 8 inches, not less than | 27 per cent. |
| Sulphur, not to exceed | 0.35 per cent. |
| Phosphorus, not to exceed | 0.30 per cent. |

The plates are to be free from laminations and surface defects, and also to stand cold and quench bending tests flat down without showing a sign of fracture. The tests are to be made by (name of testing laboratory) at the expense of the purchaser.

*Joints.* — The circular joints are to be single riveted and lapped. The longitudinal joints are to be butted and provided with inside and outside covering plates. The inside and outside covering plates are to be double riveted each side of the center of the joint, and the inside plate will be extended sufficiently beyond the outside on both sides of the joint to receive on each side two additional rows of rivets passing through it and the boiler shell. Thus there will be eight rows of rivets in each longitudinal joint. The outer rows of rivets will have double and triple the pitches of the other rows.

There will be one longitudinal joint in each plate. The boiler is to be made of three plates, and the joints of the two end plates will be above the water line on one side of the boiler and that of the center plate above the water line on the other side. If practicable it is preferred to have the seam near the bridge wall back of that wall, thus making the front plate wider than the others. The heads will be single riveted to the shell.

*Riveting and Holes for Rivets.* — The holes for rivets will be punched one-fourth inch small and drilled to size with all plates and covering plates in place. The riveting is to be done by a hydraulic machine. As little hand riveting as possible is to be done. The plates are to be fitted metal to metal before riveting, brought up to butt in contact, and are to be properly curved out to the ends. No filling pieces are to be used and no drifting is to be done.

*Calking.* — Calking is to be done with a round-nose tool.

*Tubes.* — The tubes are to be of lap-welded steel, and are to be beaded over the ends. A Prosser expander is to be used. The tube holes are to be rounded on each side to $\frac{1}{16}$ inch radius, and not beveled as is customary. The tubes are to be located as shown on the inclosed blue-print. (Fig. 16.)

*Manholes and Handholes.* — There are to be two manholes in each boiler, one on top and the other in the front head below the tubes. The manhole on top is to be at one end of the boiler just far enough from the head to enable a man to get in feet first in either direction. This is to have a pressed steel frame, plate, and yoke. The frame is to amply make up the area of plate cut from the shell and is to be double riveted thereto. The manhole in the head is to be located as shown on the accompanying print, is to be pressed on the plate, and to have a pressed steel plate and yoke. There will be a 4-inch-by-6-inch handhole in the back head.

*Bracing.* — Above the tubes there will be four longitudinal tie rods passing through the heads and upset at the ends, having a nut outside and inside the plates. There will be four braces from the heads to the shell riveted to the latter some 5 or 6 feet from the heads. These braces will be pinned to horizontal stiffening angles riveted to the heads. There will also be horizontal stiffening angles riveted to the heads above and below the longitudinal tie rods. Below the tubes the heads will be stiffened by several short diagonal braces riveted to heads and shell.

*Steam Nozzle.* — There will be one 7-inch steam nozzle midway between the heads of each boiler. It will be a steel casting double riveted to the shell, and drilled for $1\frac{1}{8}$-inch bolts.

*Steam Box and Dry Pipe.* — Bolted up against the under side of the steam nozzle, or screwed into it, there is to be a 5-by-5-by-7-inch tee. Into each 5-inch branch there is to be a 5-inch wrought-iron pipe, the lengths to be equal and determined by the distance from the steam nozzle to the manhole nozzle. The top of the pipe is to be about 1 inch below the shell, and each branch is to be perforated on top with twenty-five 1-inch holes. The outer ends of the pipes are to be capped and the pipes strapped up to the shell. There are to be two $\frac{1}{4}$-inch drain holes in the bottom of the steam box.

*Safety-valve Nozzle.* — There will be no safety-valve nozzle.

*Feed-pipe Nozzle and Feed Pipe.* — This will be on top of the boiler near the front head. The feed pipe will pass through it, branch to one side and

pass to the other end of the boiler, where it will discharge. The feed pipe will be 2-inch brass, iron size, and will be furnished with the boiler.

*Blow-off Pipe.* — This will be tapped into the bottom of the boiler, and will be 2½ inches in diameter.

*Safety Plug.* — This will be screwed into the back head in the customary position. A Lunkenheimer plug filled with pure Banca tin is to be used.

*Smoke Box.* — The smoke box will not be formed by the extension of the main shell but by bolting a ⅜-inch plate thereto, and having a smoke nozzle riveted to it. The size of the smoke nozzle will be 18 inches by 5 feet 6 inches. The smoke boxes will overhang. The smoke-box head will be of cast iron secured to the shell air-tight. The doors will likewise be of cast iron fitting against planed faces and having turned hinge pins and drilled holes. The doors are to be clamped to the head after locomotive practice.

*Damper.* — A damper with two plates with a rod between them is to be provided with a weighted lever and chain for its control.

*Front.* — The front is to be made of steel plate ½ inch thick.

*Grates.* — These will be furnished by the purchaser.

*Fire and Other Doors.* — Each boiler will have two large-size fire doors, with door frames, planed to match each other, and to have turned pins and drilled holes. The back cleaning door is to be clamped to the frame, after locomotive practice, to have planed joints, turned pins and drilled holes for pins. It will be so made that it can be securely built in.

*Supports.* — The boilers will have a double set of supporting brackets at each end, eight for each boiler, the back ones having rolls.

*Fittings.* — The contractor will furnish, but not attach, the following fittings for each boiler:

One 10-inch ——— company's steam gauge graduated to 300 pounds, and numbered every 20 pounds.

One 5-inch ——— company's safety valve set to blow at 127 pounds.

One Reliance water column without safety device, and provided with two water-glass fixture bosses on opposite sides at such distances apart as will accommodate an 8-inch exposed length of glass. There will be no gauge-cock bosses.

Two ——— company's heavy-pattern water-glass fixtures with best Scotch glasses, 8-inch exposed length.

Four extra glasses of proper length.

Extra-strong iron-size brass piping for water column, all cut to length and threaded, and holes drilled and tapped therefor in shell.

No gauge cocks will be used.

*Arch Bars.* — Proper arch bars are to be supplied with each boiler.

*Buck Stays and Rods.* — These are to be furnished and the buck stays are to be of 8-inch I beams.

*Drawing.* — The party receiving the contract is to furnish a first-class drawing of the boiler as built, and of the setting therefor.

*Testing.* — Each boiler will be tested at the works of the contractor to 190 pounds per square inch with water pressure and made tight under that pressure in the presence of a representative of Dean & Main.

*Inspection.* — The boilers will be built under the inspection of Dean & Main and the Mutual Boiler Insurance Company.

*Reservation.* — The right is reserved to reject any or all bids.

*Terms of Payment.* — Proposals are to state the desired terms of payment.

*Delivery.* — The proposal is to state the guaranteed time of delivery of the three boilers.

**Specifications for Water-tube Boilers.** — An engineer's specification for water-tube boilers can only be a very general statement of what is wanted, for the reason that such boilers vary greatly in design. The engineer can, however, describe what he wants and ask each bidder to supply information enabling him to understand thoroughly what each bidder proposes to furnish.


Fig. 17. Boiler Room, Plymouth Cordage Company.

The engineer should state in his specification the number of boilers he wishes to purchase, the amount of heating and grate surface each is to contain, the location of the plant for which they are wanted, and the name of the owner or owners of the plant. If the boilers are to be erected and placed in running order by

the builder, it should be so stated. The engineer should state the kind of coal to be burned, the kind of service to be required of the boilers, and the dimensions of the chimney and flues. It is well to furnish a sketch showing the location of the boilers, flues, and chimney. If a mechanical stoker is to be used, the boilermaker should be asked to furnish a boiler setting to suit. The engineer should require each bidder to furnish detail drawings showing all parts of the boiler and setting, in such a way as to show the facilities for cleaning and inspecting the parts and the manner of making renewals. He should ask for the physical qualities of the metals used; the material of which the tubes are made; the name of the manufacturer of the different fittings used, such as the safety valve, blow-off cocks, water and steam gauges; the character of feed piping, whether iron or brass.

For certain types of water-tube boilers with straight tubes slightly inclined from the horizontal and connecting with vertical or nearly vertical headers at the front and rear of the boilers, the headers connecting with combined steam and water drums, there are usually several different arrangements of tubes that will meet a specification calling for a certain heating surface. For instance, there may be eighty 18-foot tubes so arranged that there are eight tubes in width and ten tubes in height, or there may be ten tubes wide and eight tubes high. The latter arrangement would mean a wider boiler costing more, but giving a wider and consequently larger grate that might be needed with low-grade fuels. It is probable, the heating and grate surface and coal burned being the same, that a high narrow boiler of this type will give a better efficiency than a low wide one. If the boilermaker is to supply and erect the breechings, smoke flues, chimney dampers, and damper regulator, it should be stated in the specifications. If the engineer proportions the boiler, it is hardly fair to exact a guarantee from the maker as to the efficiency of the boiler. He should, however, guarantee that the boiler will operate at 50 per cent over its rated capacity without showing signs of depreciation.

# CHAPTER IV.

## THE SELECTION OF THE TYPES OF ENGINES, DIMENSIONS OF CYLINDERS, SPEED, STEAM PRESSURE, ETC.

THE method of buying a steam engine most frequently employed by engineers is to invite bids from a number of builders upon an engine which, with a certain number of revolutions, steam pressure and back pressure, will develop a given horsepower. The cylinder dimensions are usually supposed to be looked after sufficiently when the bidders are asked to state the steam consumption they will guarantee. On opening the bids thus obtained, it is sometimes found that some offer engines with smaller cylinders than others, to do the same work. The prices are naturally different, and it occasionally happens that the purchaser does not get the engine best suited to his needs, first cost and economy of operation both being considered. The nontechnical factory manager, knowing nothing of engines, frequently makes a mistake in this matter of purchasing an engine, as he does not know the type of engine he wants and much less what its details should be. He, therefore, frequently buys something very different from what he needs. Such an individual should either engage a competent consulting engineer to invite bids for him on the type of engine best suited to the conditions, or go to one reputable engine builder, inform him of the conditions that exist, and have an engine built to suit them. There is no doubt that there are builders fully competent to study the conditions involved and advise an owner what is required. Two or more builders should not, however, be placed in competition where the cheapest bid is likely to be the deciding factor in the selection of an engine, and an incompetent person is to be the judge of the propositions offered. It could hardly be possible to educate the nontechnical purchaser of engines, but there are a number of engineers who are not steam-engineering experts that have to buy engines without the aid of expert advice; and it is hoped to give

some general information and data to assist such in determining the type of engine, approximate cylinder dimensions, etc., best suited for different situations, so that bids can be invited upon a specific machine and thus secure the benefit of a competition, where all bids are made upon the same basis. In doing this, the selection of the type of engine will first be discussed; afterward, the determination of those factors that fix the capacity and efficiency of an engine; and finally, the method of applying these factors in the general equation for determining the power of an engine so that the cylinder sizes or volumes may be calculated.

**Selection of Type.** — To most people it is manifest that it would be bad engineering to install a costly triple-expansion engine that operates with the highest economy in a plant where fuel is of little or no value. Many situations, however, require the keenest judgment to select the type of engine best adapted for the service the engine is to perform. There is a saying that any expenditure for plant is warranted that results in savings which more than pay the interest on the expenditure, the depreciation, and the cost of repairs on the plant. This is a simple proposition in itself, but

TABLE 8. — STEAM CONSUMPTION OF DIFFERENT TYPES OF ENGINES.

| Type of engine. | Pounds of steam per horse-power per hour. | Steam pressure, pounds gauge. |
|---|---|---|
| High-speed simple | 32 | 80–100 |
| High-speed compound noncondensing | 24–26 | 150–110 |
| High-speed compound condensing | 19–21 | 150–110 |
| Corliss simple noncondensing | 26 | 80–100 |
| Corliss simple condensing | 21 | 80–100 |
| Corliss compound noncondensing | 20–22 | 150–110 |
| Corliss compound condensing | 14–15 | 150–125 |
| Triple-expansion condensing | 13 | 150– |

it is not always easy to predict the saving that will follow the installation of a machine, or what the depreciation and repairs will be. As a basis for such calculations regarding steam engines, Table 8 is given, and in it is shown the steam consumption per horse-power per hour which might be expected from various types of engines with different steam pressures. The figures given are believed to be fairly accurate in a relative sense and are supposed

to represent about what would be obtained with each engine running at its most economical load, with valves and pistons in good but not the best condition. In the figures given for condensing engines, the steam used by the air pump is included. To arrive at the actual cost of power, the fuel cost and the cost of repairs, interest, depreciation, etc., should be taken into account. It can be safely assumed that with good anthracite or semibituminous coal a boiler will evaporate 8 pounds of water per pound of coal; hence the annual fuel cost of an engine will be:

$$\text{Annual fuel cost} = P \times \text{H.P.} \times h \times c \div (8 \times 2240),$$

in which,

$P$ = the pounds of steam used by the engine per horse-power per hour;

H.P. = the average horse-power developed;

$h$ = the number of hours during the year the power is being used;

$c$ = the cost of coal in dollars per ton of 2240 pounds.

To the annual fuel cost thus obtained there should be added, say 4 per cent of the cost of the engine, to cover the interest on the investment and 8 per cent for repairs, depreciation, etc. It is assumed that, when the formula is used to compare condensing engines with simple engines, the boiler feed water in the case of the condensing plant is first warmed by the exhaust steam from the engine and afterward by the steam from the auxiliaries, so that there will not be enough difference in the temperature of the feed water in the two cases to warrant taking this difference into account.

No really reliable figures as to the cost of engines can be given, as this is constantly varying, due to the condition of supply and demand and the cost of materials. If an engineer is in doubt as to the type of engine needed for a special situation, bids on the types that are to be considered should be obtained, and from the prices thus obtained a decision can be made. The increased cost of the most economical engines is partly offset by the fact that not so great a capacity in boilers is required to run them, hence the cost of the boilers should usually be considered. Generally speaking, it has been found that the greater expense of economically working engines is more than offset by the gain resulting from their use. For all steam pressure over 100 pounds with engines over 150

horse-power, the compound engine in most situations, whether it is to be operated condensing or noncondensing, will save enough in fuel cost to pay for its increased first cost and cost of attendance, repairs, etc. There are situations where this statement would not be true; for instance, when the fuel cost is exceptionally low or when the demand for exhaust steam for heating or for manufacturing purposes is in excess of the steam exhausted by the engines. It would then, it is manifest, be poor policy to buy an expensive engine for the sake of getting an economical one, when the economical use of steam in the engine is no object. Again, if an engine is only to be run occasionally, as in the case of a relay engine to a water wheel, the engine should not be as costly a one as if it were to run more frequently. In the latter case, the interest on the increased cost of an economical engine over one using more fuel might be more than the difference in the fuel cost, owing to the short time that the engine is used. If an engine runs a large part of the time only partially loaded, as it would in driving an electrical generator for street railway plants or for electric elevators where the load is widely fluctuating, it is doubtful if compound engines would pay. Generally speaking, triple-expansion engines for mill or electric work with steam pressures as low as 150 pounds have not shown that the saving due to their use is sufficient to pay for their increased cost over a compound engine. Engines used in the power plants of very large buildings should, generally speaking, be of the compound type, if over 200 horse-power. They are naturally run noncondensing. Another point to be considered is the increased cost of attendants for the most economical engines over those with simpler parts and using more fuel.

**Steam Pressure.** — The steam pressure employed in simple engines may be said, generally speaking, to vary from 80 to 120 pounds above the atmosphere, and in compounds from 100 to 150 pounds, although the latter are sometimes run with lower pressures than 100 pounds. The tendency is toward the higher pressures. High-pressure steam, especially in compound engines, is conducive to high economy, but it should not be forgotten that high steam pressures, and by that is meant pressures of 135 pounds and over, mean increased wear on the system, much trouble with steam piping if unusual care is not taken in constructing it, and an increased loss from leakage and condensation; but for all these

objections the higher economy of engines with high steam pressures will more than compensate for the drawbacks if the plant is well designed and is placed in competent hands. In electric power stations and with mill and factory engines of large power, the pressure can well be from 135 to 150 pounds. In very large central stations pressures of 175 pounds are common. High-pressure steam is of special importance where compound noncondensing engines are used. For plants for very large buildings employing competent engine attendants, a pressure of at least 120 pounds should be selected if this type of engine is to be used; and in large mills or large electric plants, where the steam plants will be in competent hands, the steam pressure with noncondensing compound engines should be as high as 150 pounds. Even with pressures as high as 125 pounds the steam piping should have unusual attention both in the design and erection.

**Rotative Speed.** — The number of revolutions an engine is to run is frequently fixed by the fact that it is to be directly connected to a dynamo which has to be run at a certain speed, or by some other condition. The rotative speed is limited by the centrifugal force developed in flywheels when in motion, by the maximum speed at which the piston should be run, and, in the case of engines with a releasing valve gear, such as the Corliss, by the tendency of the valve gear to become noisy and give trouble at too high a speed. High-speed engines of 10-inch stroke, with automatic cut-off controlled by a shaft governor, are usually run at about 325 revolutions per minute, and the rotary speed is decreased to about 150 revolutions in engines of 24 inches stroke. An engine may be run at much lower rotative speeds than those mentioned, but it would then, of course, require a proportionally larger cylinder to develop the same power, other conditions being equal. With releasing-gear engines with ordinary air dashpots, the highest rotative speed that should be employed seems to be about 100 revolutions per minute. In electric work the rotative speed is sometimes increased above 100 revolutions per minute for engines directly connected to a dynamo, but unless some special valve gear is used, permitting high speeds, the life of the engine is shortened and it is apt to be noisy and require considerable attention. The rotative speed of an engine is frequently limited by the proper speed at which the

piston should be run. The need of a high rotative speed for electric work is due to the fact that the higher the speed the less is the cost of an electric generator. In electric work the rotative speed of an engine is determined by the speed of the generator to which it is to be connected, and the speed of an engine for this service should not be fixed until information as to the speeds of standard generators is obtained.

**Piston Speed.** — Piston speed is usually defined as being the number of feet that the piston of an engine travels in a minute's time. It is obtained by multiplying the length of the stroke in feet by twice the number of revolutions per minute. Good practice with high-speed engines limits the piston speed of small engines with a length of stroke of 12 inches to about 550 feet per minute, and with engines with a 2-foot stroke to about 600 feet. In larger engines 700 feet per minute is allowable, and in electric service this figure is sometimes exceeded in long-stroke engines to obtain a high rotative speed. It is not well to go above 800 feet, however, for, when this piston speed is exceeded, very large ports are necessary to admit the larger amount of steam required with high piston speeds, and these large ports increase the clearance of the engine and make it less economical in the use of steam.

**Mean Effective Pressure.** — When steam is admitted to a cylinder during a portion of the stroke of an engine and then further supply is "cut off," the steam, after "cut-off" occurs, begins to expand as the piston advances and consequently the pressure falls. The work done in the cylinder is proportional to the average effective pressure throughout the stroke, hence it is necessary in proportioning cylinders to know what this average or mean effective pressure should be with different steam pressures. The mean effective pressure is one of the factors in the general equation for determining the horse-power developed by an engine. Its selection is an important matter, for on it, more, perhaps, than on anything else, the economy of the engine is dependent. Theoretically, the greatest amount of work is obtained from steam when it is admitted to a cylinder up to such a point in the stroke that it will afterward expand to the pressure that the engine is exhausting against. Practically, it is not well to have so complete an expansion. The more complete the expansion is (that is, the number of times the volume of steam at cut-off is expanded),

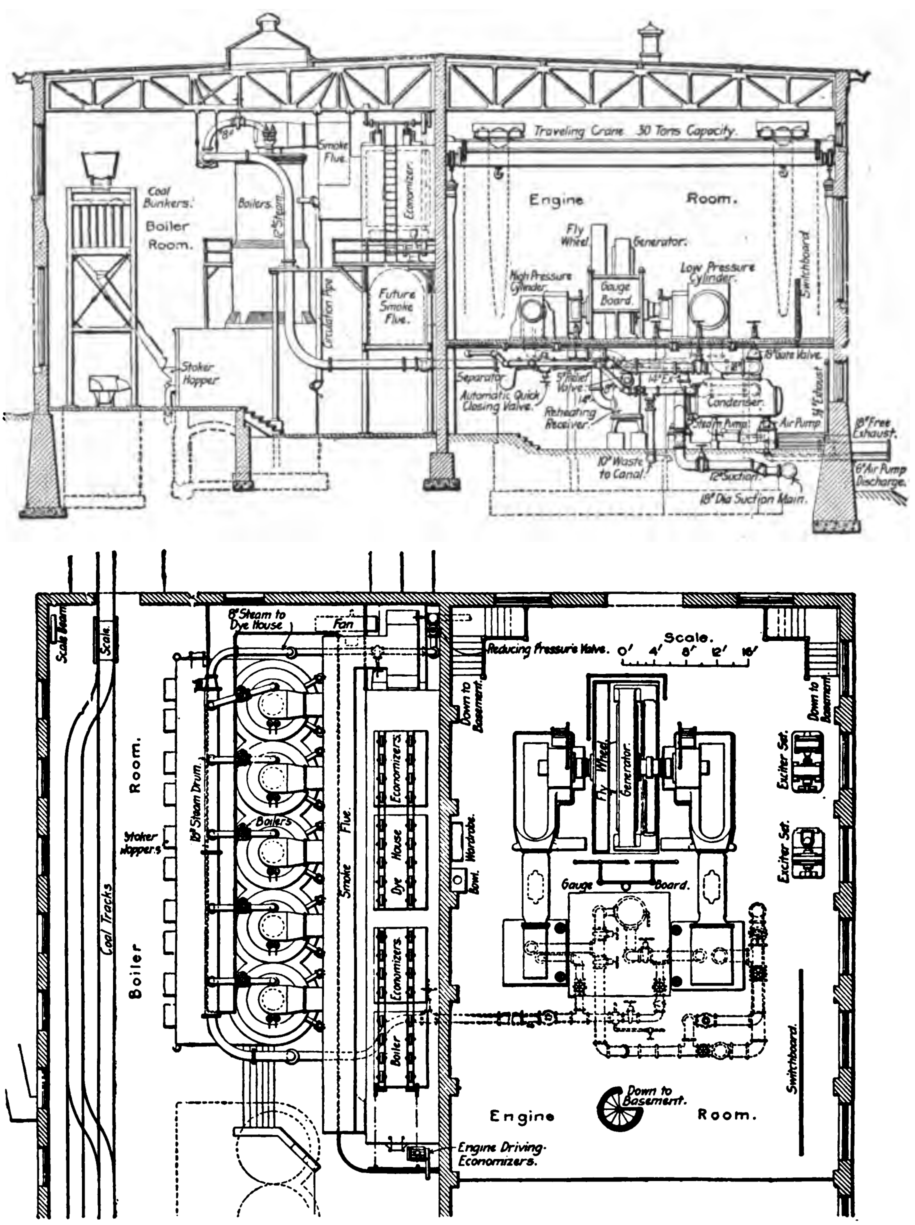



Fig. 18. Power Plant, Lancaster Mills, Clinton, Mass. (Lockwood, Greene & Co., Engineers.)

the less the mean forward pressure must be, hence engines operating with high economy work, up to a certain limit, with a lower mean effective pressure than a similar engine working with a poorer economy. The lower the mean effective pressure is, however, the larger the cylinder must be to accomplish the same work, hence the expansion may be so great and the mean effective pressure so small that the saving in fuel due to this complete expansion may not pay for the increased cost of the large cylinder. This is, in brief, the commercial problem involved in the selection of cylinder sizes.

**Mean Effective Pressures for Simple Engines.** — In Table 9 there are given the approximate mean effective pressures that are usually obtained with various steam pressures with Corliss and

TABLE 9. — MEAN EFFECTIVE PRESSURES FOR DIFFERENT STEAM PRESSURES, IN POUNDS PER SQUARE INCH, FOR ENGINES OF THE SIMPLE TYPE.

| | | | |
|---|---|---|---|
| Steam pressure, gauge | 80 | 90 | 100 |
| Corliss,* condensing | 26 | 28 | 30 |
| Corliss,* noncondensing | 36 | 38 | 40 |
| Single-valve, noncondensing | 42 | 46 | 50 |

* These data can be applied to four-valve moderate- or slow-speed engines.

other four-valve slow- or medium-speed engines, condensing and noncondensing, when the minimum amount of steam per horsepower per hour is consumed. The table also shows the same data for simple single-valve engines of the high-speed type. The overload capacities of engines proportioned in accordance with the figures given in the table are described in a later paragraph.

**Proportioning Cylinders of Single-cylinder Corliss Engines.** — The first thing to do in determining the size of cylinder for an engine to develop a given horse-power is to fix the steam pressure and afterward select a mean effective pressure proper for the conditions under which the engine is to work. Having the above data, the method of determining the cylinder size will be explained, in one case where the number of revolutions is fixed, and in another case where it is not. It will first be supposed that the number of revolutions have been fixed.

The general equation by which the horse-power of a double-acting engine is determined is in the form:

$$\text{H.P.} = P \times l \times a \times n \div 16{,}500, \quad . \quad . \quad . \quad (2)$$

in which

$P$ = the mean effective pressure in pounds per square inch;
$l$ = the length of stroke in feet;
$a$ = the mean area of the piston in square inches;
$n$ = the number of revolutions per minute.

In equation (2) the terms H.P. (the number of horse-power to be developed), $P$, and $n$ are assumed to be known, and transposing we have

$$l \times a = \text{H.P.} \times 16{,}500 \div (P \times n). \quad . \quad . \quad . \quad (3)$$

Substituting the value of $P$, $n$, and H.P., we can find the numerical value of $l \times a$. Both of these quantities are to be determined. Table 3 shows in columns 1 and 2 the diameter and length in inches of cylinders for Corliss engines, for which practically all large engine builders carry patterns. The cylinder sizes given are taken from the catalogue of the Allis-Chalmers Company. Column 3 shows the number of revolutions at which it is recommended these engines should run. Column 4 shows the product of the cylinder area in square inches by the length of stroke in feet for each cylinder, and column 5 shows the quantities given in column 4 multiplied by the number of revolutions given in column 3. Going back to the calculations, the numerical value of $l \times a$, it will be remembered, has been found. We now look down column 4 until we find a number nearest to the numerical value of $l \times a$. Opposite the number thus found are the dimensions of the cylinder that is seemingly required. One more operation remains. The length of stroke of this cylinder in feet should be multiplied by twice the number of revolutions that the engine is to run to obtain the piston speed; and if this exceeds good practice, as described in an earlier paragraph, a cylinder with a shorter stroke and larger diameter, but of approximately the same volume, can probably be found in the table, which will give the power without too great a piston speed.

If the revolutions are fixed in the first place, the problem is simpler. We then can transpose equation (2) to

$$l \times a \times n = \text{H.P.} \times 16{,}500 \div P, \quad . \quad . \quad . \quad (4)$$

TABLE 10. — DIMENSIONS OF CYLINDERS AND SPEEDS OF CORLISS ENGINES.

| Diameter cylinder, inches. | Length stroke, inches. | Revolutions per minute. | Area of cylinder in square inches × stroke in feet $= l \times a$. | Area of cylinder in square inches × stroke in feet × revolutions $= l \times a \times n$. |
|---|---|---|---|---|
| 12 | 30 | 90 | 282 | 25,447 |
| 12 | 36 | 85 | 339 | 28,840 |
| 14 | 36 | 85 | 461 | 39,252 |
| 14 | 42 | 82 | 538 | 44,177 |
| 16 | 36 | 82 | 603 | 49,461 |
| 16 | 42 | 78 | 703 | 54,889 |
| 18 | 36 | 80 | 763 | 61,070 |
| 18 | 42 | 78 | 890 | 69,467 |
| 18 | 48 | 75 | 1,017 | 76,338 |
| 20 | 42 | 75 | 1,099 | 82,467 |
| 20 | 48 | 72 | 1,256 | 90,478 |
| 20 | 60 | 75 | 1,570 | 117,810 |
| 22 | 42 | 75 | 1,330 | 99,784 |
| 22 | 48 | 72 | 1,520 | 109,477 |
| 22 | 60 | 65 | 1,900 | 123,542 |
| 24 | 48 | 70 | 1,809 | 126,669 |
| 24 | 60 | 65 | 2,261 | 147,026 |
| 26 | 48 | 70 | 2,123 | 148,660 |
| 26 | 60 | 65 | 2,654 | 172,552 |
| 28 | 48 | 68 | 2,463 | 167,484 |
| 28 | 60 | 65 | 3,078 | 200,118 |
| 30 | 48 | 68 | 2,827 | 192,265 |
| 30 | 60 | 62 | 3,534 | 219,126 |
| 30 | 72 | 55 | 4,241 | 233,263 |
| 32 | 48 | 65 | 3,217 | 209,106 |
| 32 | 60 | 62 | 4,021 | 249,317 |
| 32 | 72 | 55 | 4,825 | 265,402 |
| 34 | 48 | 65 | 3,631 | 236,058 |
| 34 | 60 | 62 | 4,539 | 281,455 |
| 34 | 72 | 55 | 5,447 | 299,613 |
| 36 | 48 | 72 | 4,071 | 293,146 |
| 36 | 60 | 62 | 5,089 | 315,539 |
| 36 | 72 | 55 | 6,107 | 335,897 |
| 38 | 60 | 60 | 5,670 | 340,233 |
| 40 | 48 | 70 | 5,026 | 351,859 |
| 40 | 60 | 62 | 6,283 | 389,558 |
| 40 | 72 | 55 | 7,539 | 414,691 |
| 40 | 84 | 50 | 8,796 | 439,824 |
| 42 | 48 | 70 | 5,541 | 387,923 |
| 42 | 60 | 62 | 6,927 | 429,486 |
| 42 | 72 | 55 | 8,312 | 457,195 |
| 44 | 48 | 70 | 6,082 | 425,748 |
| 44 | 60 | 62 | 7,602 | 471,364 |
| 44 | 72 | 55 | 9,123 | 501,774 |
| 46 | 60 | 62 | 8,309 | 515,189 |
| 46 | 72 | 55 | 9,971 | 548,427 |
| 48 | 60 | 62 | 9,047 | 560,963 |
| 48 | 72 | 55 | 10,857 | 597,154 |

and after obtaining the numerical value of $l \times a \times n$, the proper diameter and stroke of cylinder and rotative speed of the engine required can be found opposite the number in column 5 of the table nearest to the numerical value of $l \times a \times n$. If there is not a close agreement between one of the numbers in column 5 and the value of $l \times a \times n$, the revolutions may be increased or decreased as the former is respectively greater or less than the latter.

It was explained that Table 10 gives the cylinder sizes of Corliss engines made by one company and that many builders carry patterns for these sizes. Most builders have additional

TABLE 11. — DIMENSIONS OF CYLINDERS AND SPEED FOR HIGH-SPEED AUTOMATIC CUT-OFF ENGINES.

| Diameter cylinder, inches. | Length stroke, inches. | Revolutions per minute. | Area of cylinder in square inches × stroke in feet $= l \times a$. | Area of cylinder in square inches × stroke in feet × revolutions $= l \times a \times n$. |
|---|---|---|---|---|
| 9 | 10 | 325 | 53 | 17,225 |
| 10 | 10 | 325 | 65 | 21,125 |
| 11 | 10 | 325 | 79 | 25,675 |
| 11 | 12 | 300 | 95 | 28,500 |
| 12 | 12 | 300 | 113 | 33,900 |
| 13 | 12 | 300 | 132 | 39,600 |
| 14 | 12 | 300 | 153 | 45,900 |
| 14 | 14 | 275 | 178 | 48,950 |
| 15 | 14 | 275 | 205 | 56,375 |
| 16 | 16 | 250 | 268 | 67,000 |
| 17 | 16 | 250 | 302 | 75,500 |
| 18 | 16 | 250 | 338 | 84,500 |
| 18 | 18 | 225 | 381 | 85,725 |
| 20 | 18 | 225 | 471 | 105,975 |
| 20 | 20 | 200 | 523 | 104,600 |

patterns, varying more or less in size from those given, and it may be that for a particular case a builder might furnish a cylinder with slightly shorter stroke, and correspondingly greater area of piston, so that the cylinder volume would be the same. Such a cylinder would give the same power, and, because of the shorter stroke, the engine would be less expensive to build, the frame, guides, and all reciprocating parts being shorter. For this reason it is not well for the engineer to be too rigid in fixing the cylinder dimensions. Cylinders of different engines to do the same work should have the same volume; but the stroke should not be shortened too much.

**Proportioning Cylinders for Single-cylinder High-speed Engines.** — Table 11 contains the sizes of cylinders of a line of high-speed automatic engines. Although no one builder carries patterns for them all, most builders have patterns for cylinders of approximately the same volume as those given in the table. The engineer can determine the size cylinder required in the manner described for Corliss engines, using the mean effective pressure and rotative speed proper for this type of engine. A specification for this type of engine can name the length and diameter of cylinder wanted, but it should also state that bids upon engines of approximately the same cylinder volume will be considered. It is, of course, unfair to handicap a builder not possessing patterns of exactly the dimensions desired by demanding that he make a pattern to conform exactly to the engineer's specification.

**Proportioning Cylinders for Medium-speed Engines.** — It is impossible to give a table that is representative of the cylinder sizes of medium-speed engines, as those four-valve engines which run at higher rotative speed than Corliss engines and at lower rotative speed than the so-called high-speed engines are called. The reason for this is that no two builders have patterns for the same sizes of cylinders. It is, perhaps, just as well to select the engine size from the catalogue of one builder, using the same piston speed and mean effective pressure as for Corliss engines, and invite bids upon an engine with a cylinder or cylinders of equivalent volume. The shorter stroke should belong to the cheaper engine, other things being equal, for reasons explained above. An engine with too short a stroke in proportion to its diameter is not as economical usually, on account of greater clearances.

**Proportioning Cylinders for Compound Engines.**—The method of determining the size of cylinders of compound engines is somewhat complicated by the fact that steam has to be expanded through two cylinders. Theoretically, the size of the high-pressure cylinder for a given number of expansions in the engine has absolutely nothing to do with the capacity of the engine. In fact, a simple engine, with its single cylinder the same size as the low-pressure cylinder of a compound engine, would, theoretically, develop the same power as the compound if the cut-off in the single cylinder were adjusted so as to give the same number of

expansions as exist in the compound engine. In practice, however, the large number of expansions common in compound engines would not do in a simple engine, because of the excessive cylinder condensation that would occur. Dividing the expansion between two cylinders, as is done in the compound engine, reduces the condensation, and that is why compound engines are used. For the purpose of determining the cylinder sizes of a compound engine, it can be assumed that all of the work is done in the low-pressure cylinder, and then determine its size by the formulas given for the simple engines, assuming a mean effective pressure that is proper for a compound engine. After the size of the low-pressure cylinder is determined, the high-pressure cylinder dimensions can be found by selecting one whose volume is in the same ratio to the volume of the low-pressure cylinder, that practice has found to be proper.

**Mean Effective Pressures for Compound Engines.**—The mean effective pressures in compound engines are different from those in simple engines for the reason that the steam pressures and ratios of expansion are higher. In Table 12 there are given values that can be assumed for the mean effective pressures for

TABLE 12.—MEAN EFFECTIVE PRESSURES FOR DIFFERENT STEAM PRESSURES, IN POUNDS PER SQUARE INCH, FOR COMPOUND ENGINES.

| | | | |
|---|---|---|---|
| Steam pressure, gauge | 100 | 125 | 150 |
| Corliss,* condensing | 18 | 20 | 22 |
| Corliss,* noncondensing | 29 | 31 | 33 |
| Single-valve, high-speed, condensing | 22 | 24 | 26 |
| Single-valve, high-speed, noncondensing | 32 | 34 | 36 |

* These data can be applied to four-valve moderate- or slow-speed engines.

substitution in equations (3) and (4) to find the size of the low-pressure cylinder of compound engines of various types. The mean effective pressures given are equivalent to the mean effective pressures actually found in the low-pressure cylinder added to the mean effective pressure in the high-pressure cylinder multiplied by the ratio of high-pressure to the low-pressure piston areas. The mean effective pressures given for compound engines are believed to be such as will insure the lowest steam consumption per horse-power. Overload capacities are discussed later.

The data given in the tables have been gathered mainly from an examination of a number of tests made by engineers known to obtain trustworthy results, where the conditions have been such as to obtain the highest efficiency. In all cases the mean effective pressure is based upon a back pressure of 16 pounds absolute in the case of noncondensing engines, and a vacuum equivalent to 26 inches of mercury in the case of condensing engines. After the mean effective pressure is decided on, formula (3) should be used if the revolutions are not fixed, and formula (4) if they are determined upon, just as was done with simple engines, in calculating the area and length of stroke of the low-pressure cylinder of a compound engine.

After the size of the low-pressure cylinder is determined, the high pressure can be found by selecting one whose area is in the same ratio to the area of the low-pressure cylinder, as practice has found to be proper, as has been said. There is a difference of opinion among engineers as to the proper ratio of cylinder volumes with compound engines. Some engineers proportion the cylinders so that the amount of work done in each will be the same. This results, particularly with high pressures, in a considerably greater range of temperature in the high-pressure than in the low-pressure cylinder; and by many this is held to be bad practice, for it is well known that too great a temperature range results in excessive cylinder condensation, and it is to overcome this enemy to high economy in the steam engine that compounding is resorted to. The general practice with Corliss condensing engines, running with constant loads, has been to use a ratio of 1:3, 1:3½, and 1:4 with steam pressures of 125, 135, and 150 pounds respectively. Recently the tendency of a few engineers and builders has been toward a comparatively larger low-pressure cylinder than is given. Mr. George I. Rockwood, M. Am. Soc. M. E., has persistently advocated a cylinder ratio as high as 1:7 for a steam pressure of 150 pounds; and the most economical compound engine ever tested, that at the Grosvenordale Mills, Grosvenordale, Conn., which gave a steam consumption under 12 pounds per horse-power per hour, had a cylinder ratio of about that proportion. A number of other engines with similar cylinder ratios have shown almost as good an efficiency. With high-speed automatic single-valve engines the cylinder ratio varies from about 1:2½ with 100 pounds pressure to about 1:3

with a pressure of 150 pounds. An advantage in having the high-pressure cylinder fairly large in comparison with the low-pressure is the greater capacity for overloads that the engine will then have.

**Engines with Variable Loads and Overload Capacities.** — The mean effective pressures that are given for both simple and compound engines assume that the engines are to run with a steady load, and that it is desirable that they should work with the highest economy. In other words, the mean effective pressures given are such as will secure the lowest steam consumption. The matter of maximum economy is often of less importance in an engine than its maximum capacity, to take care of overloads for short periods. All engines are able only to work at maximum economy at a certain load. As the load diminishes or increases, the steam consumed per horse-power developed becomes greater. With a variable load an engine has to be large enough to supply the maximum power required, but when the load varies it is, generally speaking, better to proportion the engine so that it will be operating at the highest economy when working against a load

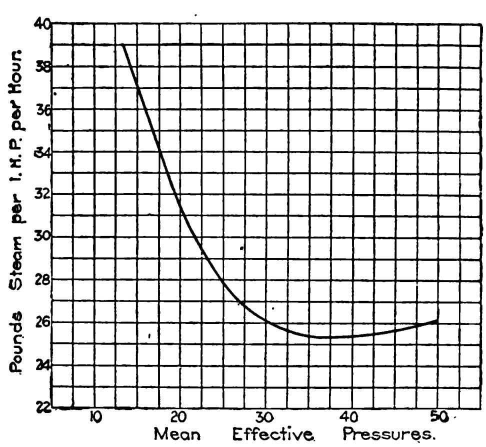



Fig. 19. Economy Curve, Simple Corliss Engine.

that is something less than the maximum that it will be called upon to supply, for the reason that it will be working most of the time partly loaded, and it should be proportioned to meet this average load with the consumption of as little steam as possible. An engine should be of such a size that its most economical load,

that is, the load under which it will operate with the least steam per horse-power, is equal to the average power required, provided the maximum power that it can develop is not less than the maximum power that will be required. Hence, if an engine is to furnish power where the maximum demand is 400 horse-power and the average demand 300 horse-power, the engine required would be one whose most economical load was 300 horse-power, and 300 horse-power would be the quantity substituted in the general equations (3) or (4) for determining the cylinder sizes.

For the purpose of showing the variation in steam consumption due to a variation in load, the results of a number of tests, believed to be reliable, made upon engines of various types, are shown in the accompanying diagrams. Fig. 19 shows the result of a test by Mr. George H. Barrus upon a 16-by-42-inch simple noncondensing Harris-Corliss engine running 85 revolutions per minute with a steam pressure of 100 pounds. It will be noticed that the most economical rating on the basis of steam consumed per indicated horse-power per hour was when the mean effective pressure was about 40 pounds, increasing slightly at 47 pounds, which was probably very near the maximum load of the engine. If, therefore, an engine is rated in accordance with the mean effective pressures given in Table 9 it is probable that the simple noncondensing Corliss engines, with a single eccentric driving both the steam inlet and the exhaust valves, would stand an overload of at least 25 per cent, and simple condensing Corliss engines over 50 per cent.

A number of tests made upon simple high-speed engines, with single valves, running noncondensing, are reproduced in Fig. 20, and the curves there shown representing the steam consumption with the variation in the mean effective pressure are numbered to correspond with the data in Table 13. Tests numbered 1 and 2 were made by Mr. J. M. Whitham and were printed in the *Engineering Record* of July 9, 1898. Test number 4 was made by Prof. R. C. Carpenter and was taken from Paper DXXIII, Transactions American Society of Mechanical Engineers. The remaining tests were made by Mr. E. J. Armstrong and were taken with his permission from a paper read by him before the Engine Builders' Association. Test number 2 was made upon a very small engine, but the test was not carried far enough to give exact data as to the proper load for it. It is

TABLE 13. — DATA OF SIMPLE NONCONDENSING HIGH-SPEED ENGINE TESTS SHOWN IN FIG. 20.

| Number test. | Diameter cylinder. | Stroke cylinder. | Revolutions per minute. | Steam pressure. | Make of engine. |
|---|---|---|---|---|---|
| 1 | 13 | 12 | 280 | 100 | Ames |
| 2 | 8 | 10 | 350 | 100 | Ames |
| 3 | 13 | 12 | 250 | 95 | Ames |
| 4 | 12 | 14 | 245 | 80 | McEwen |
| 5 | 13 | 12 | 280 | 95 | Ames |
| 6 | 13 | 12 | 250 | 95 | Ames |
| 7 | 13 | 12 | 250 | 95 | Ames |
| 8 | 17 | 16 | 225 | 100 | Ames |
| 9 | 17 | 16 | 270 | 100 | Ames |

interesting as showing the influence of size upon the steam consumption. Curves 8 and 9 were obtained from an engine of the same make, only considerably larger, and show a better economy

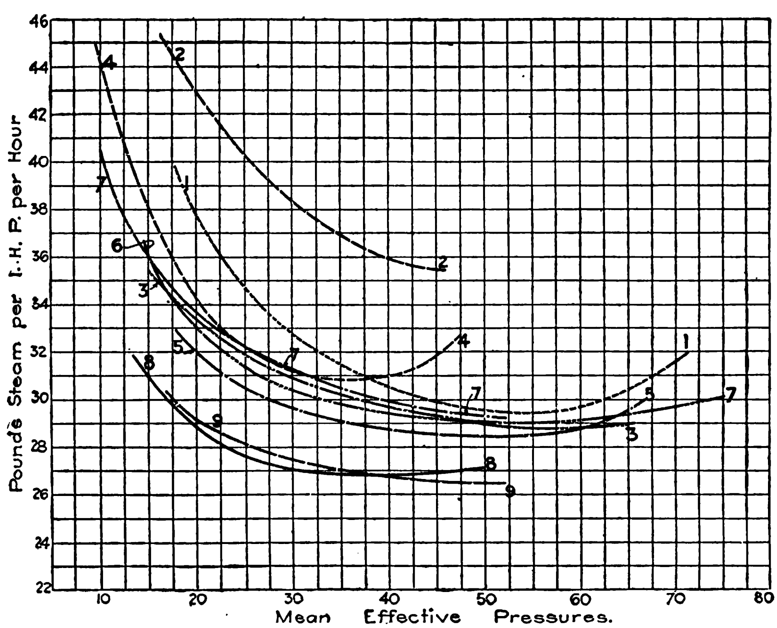
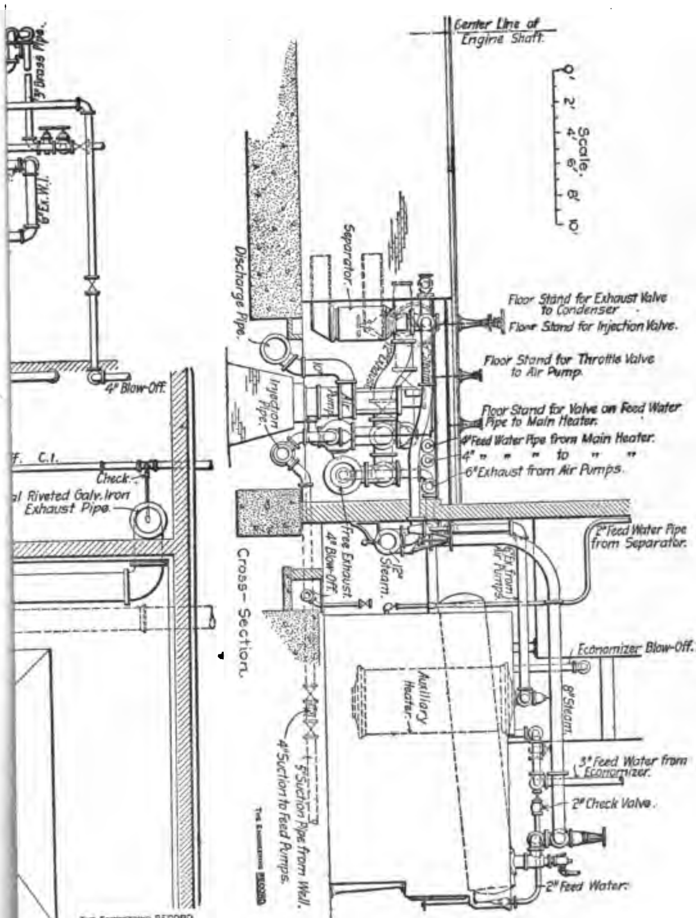



Fig. 20. Economy Curve, Single-valve Simple Engines.

due, doubtless, to the larger size of engine. Curve number 4 shows the steam consumption of an engine with a boiler pressure of only 80 pounds above the atmosphere. The most economical mean effective pressure in the latter case appears to be a little



Cross-Section.

URN, MASS.

less than half the boiler pressure. Most of the remaining curves show the steam consumption of engines operating under pressures of from 95 to 100 pounds. The most economical result with a steam pressure of 95 to 100 pounds seems to be with a mean effective pressure of from 45 to 55 pounds. Generally speaking, therefore, with the mean effective pressures recommended in Table 9, the curves, with the exception of those numbered 2, 8, and 9, which were not carried out to the maximum capacity of the engine, seem to indicate that overloads of about 33⅓ per cent could be met without a reduction in the speed of the engine. Data upon single-valve high-speed engines running condensing were not obtainable, as they are not often used.

Curves 1 and 2, Fig. 21, show the variation in economy of a 9 and 16-inch by 15-inch McEwen compound single-valve engine operating under a steam pressure of 112 pounds, a vacuum of 22 inches, and a rotative speed of 265 r.p.m. One test was made with the cylinders steam-jacketed and one with the jackets shut off. The tests were made by Prof. R. C. Carpenter and have been taken from Volume DXXIII, Transactions of the American Society of Mechanical Engineers. It is seldom that engines of this type are provided with steam jackets, as the general impression seems to be that the saving does not warrant the expense. It will be noticed from the unjacketed test that the most economical rating, with a steam pressure of about 110 pounds, seems to be with a mean effective pressure, referred to the low-pressure cylinder, of 26 pounds. Upon this basis it would seem that a compound condensing high-speed engine could easily run with an overload of over 50 per cent, particularly if the automatic cut-off be applied to both cylinders.

Curve number 3 in Fig. 21 shows the variation in economy of a 12 and 20-inch by 13-inch noncondensing engine, made by the Ball Engine Company and tested by Messrs. George H. Barrus and W. S. Monroe. The average boiler pressure was about 166 pounds, with practically no back pressure except that of the atmosphere. The rotative speed was about 175 r.p.m. The most economical mean effective pressure seems to be about 40 pounds compared with about 26 pounds for the unjacketed test of the McEwen condensing engine. While there is considerable difference in the steam pressure used by the two engines, it will be noticed that, if the loads were the same, the noncon-

densing engine would require a smaller cylinder than the condensing engine, provided both were operating with the lowest possible steam consumption per unit of power developed. Had the Ball engine been rated on the basis of a mean effective pressure of 40 pounds referred to the low-pressure cylinder, the overload capacity would have been about 33⅓ per cent. The tests were not carried sufficiently far, however, to determine if the engine could be operated at greater load than that shown by the chart.

The curve shown in Fig. 22 gives the results of tests made by Messrs. D. C. and W. B. Jackson upon a 700-H.P. Allis-Chalmers cross-compound vertical Corliss engine operated noncondensing and direct connected to a 500-kw. electrical generator. The engine has a high-pressure cylinder 24 inches and a low-pressure cylinder 40 inches in diameter, both with a stroke of 36 inches. The speed is 100 r.p.m. and the steam pressure 150 pounds gauge pressure. The steam consumption is based upon brake horse-power, so that the steam consumption per indicated horsepower per hour is close to 20 pounds. An exceedingly interesting feature brought out by the tests is the effect that the varying of the back pressure has upon the economy.

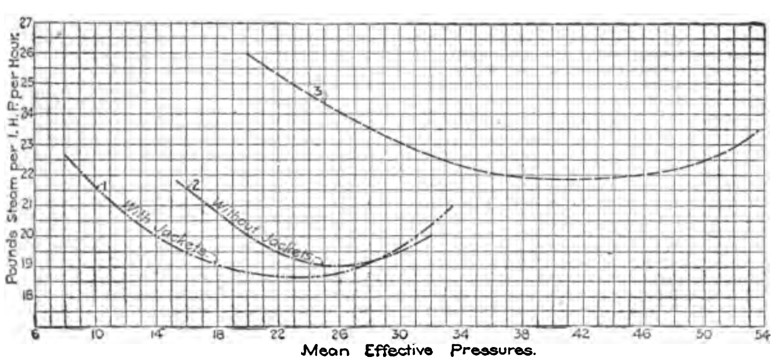



Fig. 21. Economy Curve, Single-valve Compound Engines.

The compound noncondensing Corliss engine rated upon the mean effective pressures given in Table 5 can easily stand 25 per cent overload if there is but a single eccentric driving the valve gear, and more than that if separate eccentrics are used to operate the steam and exhaust valves, for with two eccentrics a much later cut-off in the cylinder can be secured than with one. With compound condensing Corliss engines rated upon the

mean effective pressures given in Table 5, overloads of at least 50 per cent with one eccentric and probably 75 per cent with two eccentrics could be withstood without very greatly affecting the engine's speed.

**Superheated Steam, Steam Jackets, and Reheaters.** — Superheated steam, steam jackets inclosing the walls of steam cylinders, and reheating receivers placed between the cylinders of multiple-expansion engines are used to reduce cylinder condensation. The almost universal use of superheated steam on the continent of Europe in steam plants of recent construction and the remarkably high economy secured, gains from 10 to 20 per cent over the use of saturated steam having been reported, have drawn the attention of American engineers to this practice, so that the use of superheated steam is increasing considerably in the United States. The practical objection to its use, heretofore, has been in the deterioration of the superheating devices used and in the difficulty of lubricating engines where the steam is highly superheated. Recently three or four makes of superheaters have been introduced in the United States, and their extensive use abroad has demonstrated their durability and efficiency. The use of poppet valves in steam engines has overcome the difficulty of lubrication, although for temperatures under 500° F. these are said to be unnecessary. Superheaters are of two types, — those that are placed in the path of the gases in the boilers, or in the flue leading from the boiler to the chimney, and those that are heated by an independent furnace. Superheated steam is particularly adapted to steam turbines because of the increase in economy and the diminished wear of the turbine buckets or blades, which is greater with saturated steam, due, it is believed, to the erosive effect of the entrained moisture.

Steam jackets are seldom used upon any engines but those of the slow and medium rotative-speed types, and their use becomes of less value as the speed is increased. Except for slow-moving pumping engines their value is still a matter of doubt, although some engine builders provide them. Reheating receivers are in more common use, but their value is also a matter of dispute. The heating is done by a coil filled with steam of a higher pressure than that of the steam passing through the reheater. Compound- and triple-expansion engines of the slow- and medium-speed type are usually provided with them.

# CHAPTER V.

## SPECIFICATIONS FOR STEAM ENGINES.

THERE is occasionally a tendency on the part of some to ridicule elaborate specifications for steam engines, yet as a specification is a description of what one party to a contract agrees to furnish to another, it should be sufficiently complete to define exactly what is to be supplied. A complete specification is unnecessary, perhaps, where an owner engages a builder, without competition, to construct and install an engine suited for existing conditions and agrees to pay for whatever the builder elects to supply. In competitive bidding, the builder is not legally bound to supply anything more than that which the specification calls for, and all bidders are not apt to figure on doing more than that in preparing their bids, knowing full well that other bidders will not do so. Hence, if it is found, after a contract is made, that there are some desirable details which have been overlooked by the engineer in his specification, they must be purchased and paid for, as extras, at, naturally, a higher figure than what they would cost if specified in the beginning.

The amount of detail necessary in a specification naturally varies with the size of the engine to be purchased and the extent to which the design departs from the standard types for the service. It is the author's intention to call attention to a number of details which must be considered in buying a large engine, although reference to them may be omitted in a specification for less important work. Some engineers believe in specifying the cylinder dimensions, at least in a general way, so that all bidders may bid upon engines of equal capacity. Methods of determining these dimensions for simple and compound engines of various types were given in the preceding chapter. While they may be fixed by the engineer, some latitude should be given bidders in order that standard patterns may be utilized. In those situations where an engineer can control the purchase of an engine, as they can in private work, it is safer to specify the conditions and

requirements and allow the engine builders to bid on cylinder dimensions of their own choosing, and this course is usually followed.

A specification usually begins by stating the type of engine or engines wanted, whether it is to be of the simple, compound, or triple-expansion type, whether it is to be run condensing or noncondensing, and where it is to be located. Even though the engineer fixes those dimensions that determine the capacity of the engine, the specification should state the load at which the engine is to operate with the highest economy, the maximum load that it is to be called upon to operate, and the kind of service to which it is to be subjected. If guarantees as to capacity and efficiency are to be required from the builder, these data are, of course, essential. Even if they are not, and the engineer assumes the responsibility for the fulfillment, by the engine, of the requirements imposed by existing conditions, as he does when he fixes the cylinder dimensions, steam pressure, and rotative speed, the engine builder should be informed as to what will be required of the engine, as he is very apt to make calculations that will serve as a check on those of the engineer.

Sometimes a specification states that the price of the engine is to include its delivery and erection on a foundation supplied by the owner, and placing it in proper running condition. Again, however, the engine is sold free on board cars at the railway point nearest the locality where the engine is to be used. In the latter case the builder usually supplies a man to take charge of the erecting of the engine, which is done by labor employed by the owner under direction of the builder's erector.

If the engineer has determined the cylinder dimensions, the rotative speed, steam pressure, and whether the engine is to be run condensing or noncondensing, the specification should give: Diameter and stroke of cylinders; number of revolutions per minute; horse-power to be developed when working at highest efficiency; horse-power at maximum load; steam pressure; back pressure if run noncondensing, or vacuum if run condensing. If the engineer does not fix the cylinder dimensions, he should give the rotative speed, steam pressure, and other data above mentioned and ask that each bidder state the dimensions of the cylinder they propose to supply.

As before stated, an engineer should not be too rigid in regard

to cylinder dimensions, and it is well, particularly when purchasing high-speed or medium-speed engines, to state in the specifications that engines with cylinder volumes equivalent to those specified will be considered. There are objections to an engine with too short a stroke in proportion to the area of piston; these were stated in the preceding chapter. If the engine is to be run condensing and the builder is to supply the condenser, provision should be made for it.

The engineer should describe in a general way the kind of valve gear that is wanted. If the engine is to be of the compound Corliss type, the automatic cut-off might be applied to one or both cylinders. It is better to have it act on both cylinders if the load is variable, for the reason that a more equal distribution of the load between the cylinders will occur at light loads than there will if the cut-off is applied only to the high-pressure cylinder. Again, for Corliss engines, a separate wrist plate for the steam and exhaust valves, each driven by a separate eccentric, may be specified so that the engine may work with a later cut-off and consequently greater overload than is possible if both valves are driven from one eccentric. The cut-off may be applied to both cylinders in other types of compound engines.

If the cylinders are to be steam-jacketed in barrels or heads, or both, it should be so stated. Consulting engineers seldom pay much attention to this question, as the value of steam jackets is still a disputed point. Generally, engines for electric service and factory work are not steam-jacketed, particularly if the piston speed is over 600 feet per minute. It is the custom in large multicylinder engines of the medium-speed and slow-speed types, to place reheating receivers in the steam pipes between the cylinders, in which the steam is heated in transit from one cylinder to the other by live steam admitted to a coil of pipe. The value of these reheaters is also a matter of dispute. However, if they are wanted, they should be specified, also the manner in which the condensation that occurs in them is to be disposed of. The steam passing through the heater from one cylinder to the other is usually at a much lower pressure than the steam in the coils, hence the condensation must be drawn off by separate traps. The connection of these traps with the reheater and with a receptacle into which they can discharge should be made either by the engine builder or the steam-piping contractor.

If the engine is to be of the Corliss type, the kind of bed that is wanted should be mentioned. Two forms are made by most builders; one, known as the heavy-duty type, is adapted for heavy work and high pressures, and the other, the girder frame, is lighter and is of older design. The heavy-duty design is almost

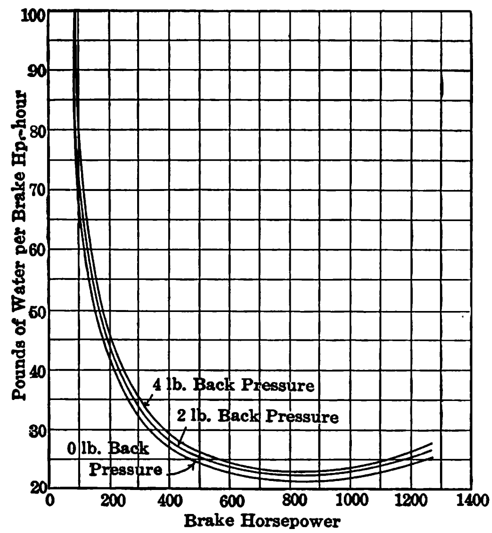



Fig. 22. Economy Curves, Corliss Compound Noncondensing Engine.

invariably used for pressures over 125 pounds and to an increasing extent in lower pressures; being much stronger, it is a little more expensive. If bids are called for a high rotative-speed automatic engine, the specifications should call for an iron sub-base with the engine, for all builders do not furnish them except at extra expense. Occasionally the iron sub-base is not used and the bed of the engine is bolted to a brick foundation which is built up above the floor the height of the sub-base.

There are a number of points concerning the design of the engine, chiefly relating to dimensions of wearing surfaces and the strength of parts, that engineers rarely fix. It is the custom of some, though, to ask each builder what he intends to furnish in respect to certain details, usually about as follows: Diameter and length of bearings, diameter and length of crosshead pins, diameter and length of crank pins, diameter of shaft in the body, dimensions of crosshead shoes, length of connecting rod, diam-

eter and face of flywheel (both may be fixed by the engineer), weight of flywheel, weight of engine bed, weight of entire engine.

When these data are received from each builder the engineer can tabulate them and compare the proportions of the engines offered. If any part of an engine differs from that of others enough to warrant such a course, the engineer can ask the reason for this deviation, and if it is not a good one the builder can be requested to modify his design, or the bid can be rejected. The engine builder should be asked to furnish a blue-print or drawing of some kind showing a plan and elevation and the principal dimensions of the engine he proposes to furnish.

Regarding details of construction of engines, most American engines are built after standard designs adopted by each builder, and not much attention is usually paid in specifications to these details when builders of established reputation are bidding. Most builders furnish catalogues containing illustrations of the details of various parts of their engines, such as the valves, valve gear, governor, piston, main bearings, crosshead, etc. If an engineer is obtaining bids on an engine the details of which are unknown to him, such illustrations can be asked for. When about to execute a contract with a builder, it can be stated in the contract that the details of construction are to correspond in design with drawings or blue-prints or catalogue sketches furnished by the builder.

There are a great many minor details of an engine, such as oil cups, sight-feed lubricators, throttle valve, relief valves on the cylinders to prevent them from being damaged by water, cylinder lagging, steam and vacuum gauges, gauge board, provision for attaching the indicators to the cylinders, etc., that have to be looked after, and many builders furnish a printed specification which refers in more or less detail to some of these matters. This specification can be asked for and made part of the contract, but before so doing care should be taken that it does not conflict with the specification issued by the engineer.

The character of materials used in engine construction is not usually given much attention in an engineer's specification, but recently the materials of which shafts of large engines are made is receiving more thought, and the kind of metal used, and its treatment, is frequently specified. The most advanced practice is found, probably, in the fluid-compressed hollow-forged treat-

ment, perfected by the Bethlehem Steel Company. The molten steel is poured into a cylindrical mold and subjected to an enormous pressure while cooling, so as to diminish the blowholes that frequently occur in the ordinary method of casting. The ingot is cooled slowly and the impurities usually collect at its axis; after it has cooled, an axial hole is bored through it to remove these impurities. The ingot is then reheated and a mandrel is slipped through this hole, after which it is forged under a slow-moving hydraulic press, the action of which penetrates much more deeply into the metal than is the case with the blow of a steam hammer. After forging, the ingot is annealed to remove internal strains and improve the structure of the metal, which is then machined to size. Sometimes the shaft is oil-tempered after annealing. Open-hearth and nickel steels are used, the small percentage of nickel added tending to raise the physical properties of the mixture. A large number of recent important engines for mill and electric power-house service have been equipped with shafts made in this way.

The time of delivery of an engine should be given in a specification, and with large engines the builder is sometimes required to furnish a man to operate the engine for a short period after it is erected. If the engine room is to be fitted with a traveling crane operated by electricity or by hand, it is well to notify the engine bidders to that effect, as the use of a crane reduces the cost of erecting engines, and the owner should have the benefit of this.

In regard to steam piping, the engine builder usually supplies and connects the piping between the cylinders in a compound or triple-expansion engine. In large compound engines, particularly of the cross-compound type, that is, with the cylinders placed side by side and driving separate cranks, it is frequently the custom to place a valve in the pipe carrying the exhaust steam from the high-pressure to the low-pressure cylinder, and to run a branch from the high-pressure steam pipe to this exhaust pipe, connecting with it at a point between the low-pressure cylinder and the valve mentioned. With this arrangement the high-pressure cylinder can be cut out, and the engine driven by the low-pressure side alone. The pipe conveying high-pressure steam to the low-pressure cylinder is usually provided with a pressure-reducing valve as well as an ordinary stop valve. By connecting

the exhaust from the high-pressure cylinder, at a point between that cylinder and the stop valve between the two cylinders, with the main exhaust pipe, and by placing a valve in the exhaust from the low-pressure cylinder, the low-pressure cylinder can be entirely cut out and the engine driven by the high-pressure cylinder. The advantage of such an arrangement in the event of the breakdown of one side of the engine is obvious. An arrangement of this kind is shown very nicely in Fig. 18. This practice is more common in mill and factory work than it is in electric generating plants.

For electric work, where revolving parts of dynamos are mounted on the engine shaft, as is the custom in directly connected work, the specifications should require the engine builder to cut a key-way in the shaft. The dynamo builder usually supplies a key for keying the armature of the dynamos on the engine shaft. The dynamo and engine are sometimes shipped by their respective builders to the site of the power plant and the engine or the dynamo builder fits the armature to the shaft. In other instances the shaft is shipped to the dynamo builder and is put on the shaft by the latter. The specification should state what each builder is to do in this respect.

A specification often asks what steam consumption the builder of an engine will guarantee. If a guarantee as to steam consumption is to be made, it should be of the following general form: "The engine is guaranteed to consume not more than —— pounds of steam per indicated horse-power per hour when developing —— horse-power when running at a speed of ——— revolutions per minute with a steam pressure of —— pounds above the atmosphere, and with a back pressure of —— pounds above the atmosphere." The specification should state that in case of dispute the steam pressure is to be the average pressure as obtained by a throttled steam gauge connected to the steam pipe close to the throttle valve, or by means of a steam-pipe diagram obtained by attaching an indicator to the location named. If the engine is to run condensing instead of with a back pressure above the atmosphere, the vacuum to be carried should be stated. The usual vacuum is from 26 to 27 inches of mercury, depending upon the size of the engine. If the engine is provided with steam jackets and reheating receivers, it should be stated that the steam used by them is to be included in the consumption of

the engine. If the engine is to run condensing, the steam used by the air-pump should be included, provided the engine builder supplies and is thereby responsible for the efficiency of the condenser. Sometimes, when condensers are not supplied by the engine builder, the steam used by the air-pump is not considered as being part of the steam used by the engine. If the air-pump steam is or is not to be considered as part of the engine consumption, a statement to that effect should appear in the guarantee. It is usually the custom to measure the degree of vacuum obtained by a gauge attached to the exhaust pipe of the engine close to the engine, and a specification should state that the vacuum is to be so measured.

A fair guarantee to ask concerning the regulation is that the speed of the engine shall not vary more than $1\frac{1}{2}$ per cent above or below the normal speed under any condition of load. The builder should guarantee the workmanship and materials to be of the best, and agree to make good at his expense any defects in the engine, not due to neglect, that develop during the first six months or year it is in operation.

The accompanying specification was prepared by Mr. Nicholas S. Hill, Jr., M. E., and it is printed here with his permission. It should be understood that it was drawn for the purpose of obtaining a small automatic high-speed engine for a private corporation that obtained bids from several reliable builders whose engines were well known.

## SPECIFICATION FOR A 300-I.H.P. NONCONDENSING ENGINE.

In the following specification the noun "Company" is used to designate the purchaser, the (purchaser's name and address). The noun "Builder" is used to designate the seller, the contractor, the manufacturers of the engine.

*Rejection.* — The Company reserves the right to reject any or all bids.

*Engineer.* — The interpretation of the specifications hereinafter set forth shall be left to the Engineer appointed by the Company, and the inspection of all materials furnished and all tests for the determination of the fulfillment of the guarantee herein contained shall be made under the direction of the said Engineer.

*Test.* — The engine will be tested at such time, after the erection and completion of engine and generator, as the builders may select and after the Engineer shall have received at least one week's notice. The Company will furnish the necessary fuel, oil, and supplies, and the contractor will be required to furnish the indicator rig and prepare the engine for test. The Engineer will furnish the indicators. A run of ten hours will be made and the load will be

maintained as nearly as possible at 300 I.H.P. The conditions of the test will be fixed at the time of the signing of the contract as agreed between the Engineer and the Builder.

*Kind of Service.* — The engine is intended to operate a generator driving motor-driven tools located in the shops of the Company.

*Location.* — The engine is to be located on foundations in the power house of the (name) Company at (place).

*Type of Engine.* — The engine desired is of the single-valve, tandem compound, flywheel governor, automatic cut-off type, with extension sub-base, direct connected to a (name) generator, size No. —, 200-kw. 250-volt generator.

*Conditions of Erection.* — The Builders will furnish and erect engine on foundation supplied by the Company. The engine may be unloaded directly from car at the door of the power station within 15 feet of the foundation. The station is equipped with a traveling crane of five tons capacity. The armature for the generator will be placed on shaft at the power house.

*Conditions of Operation.* — The engine is to run noncondensing at 200 r.p.m. Steam pressure 125 pounds above the atmosphere. Back pressure 15 pounds absolute. The maximum load for which engine is intended equals 400 I.H.P. The engine is to operate at highest efficiency with load equal 300 I.H.P. The average load will equal 175-200 I.H.P.

*Size of Cylinders.* — The cylinders shall be approximately 18 inches and 29½ inches by 18 inches, or 18½ inches and 30 inches by 17 inches.

The volumes of cylinders furnished shall be not less than the volumes herein specified, and the ratio of the cylinder volumes shall be between 2.6 and 2.8.

*Speed Regulation.* — The speed regulation shall be within 1.5 per cent above or below normal.

*Piston Speed.* — The piston speed shall be not less than 560 feet per minute nor more than 600 feet.

*Clearance.* — The clearance shall not exceed 8 per cent.

*Indicator Attachment.* — Brass piping for indicator with three-way valve is to be furnished for both high- and low-pressure cylinders.

*Dimensions and Weights.* — The Builder shall furnish the following data in regard to engine:

Total weight of engine.
Total weight of heaviest part.
Total weight of flywheel.
Diameter and length of main bearings.
Diameter and length of crosshead pins.
Diameter and length of crank pins.
Diameter of main shaft in body.
Diameter and face of flywheel.
Length of connecting rod.

*Blue-prints, Template, Specifications.* — The Builder shall furnish blueprint of engine and foundations which shall form a part of these specifications after acceptance of proposal. A foundation template and foundation bolts shall be furnished by the Builder. The foundation bolts to be provided with washers at least 10 inches square and to be threaded for two nuts at top. The

Builder shall also furnish specification of the lubricators, oil cups, tools, etc., to be supplied with engine.

*Materials.* — All materials used in the construction of the engine to be the best of their various kinds and to be in strict conformity with the latest modern practice.

*Guarantee.* — The engine shall be guaranteed to consume not more than 24 pounds of steam per I.H.P. per hour, when developing 300 H.P., when running at a speed of 200 r.p.m. with a steam pressure of 125 pounds above the atmosphere and with a back pressure of 15 pounds absolute.

The Builder shall also guarantee to make good any or all defects developed, within 150 days from the time of starting engine, which may be due to inherent defects in materials or faulty workmanship and design, provided such defects are developed when engine is running at less than 50 per cent overload.

*Painting.* — All unfinished iron work about engine shall be filled, rubbed smooth, and receive one coat of paint before leaving shop. A second coat is to be applied after erection and finally a finish coat highly enameled. The color is to conform with machinery already in place in the engine room. Quality of paint to be such as to insure against blistering, peeling off, or fading, and shall be satisfactory to the Engineer.

*Covers.* — The engine is to be supplied with a neatly fitting and well-made oiled-canvas cover to be approved by the Engineer.

*Time of Delivery.* — The Builder shall specify the earliest possible date of delivery.

*Price.* — The price submitted by the Builder shall include delivery and erection in conformity with the preceding specification.

The following specification was prepared by an engineer for the purchase of Corliss engines at a public letting, where anyone could submit a tender that wanted to; hence the specifications go into details a good deal more than is necessary, perhaps, for private work, where the bidding can be confined to experienced and trustworthy builders.

## SPECIFICATIONS FOR CORLISS ENGINES.

*Intent of Specifications.* — The intention of these specifications is to provide for furnishing and installing, complete and ready for operation in a manner satisfactory to the engineer, three simple Corliss type engines, each to be as hereinafter specified, for direct connection to direct-current generators, and certain other apparatus, all as hereinafter specified, in the power plant for —— at ——. The owner will provide a 20-ton hand-operated traveling crane.

*Time of Delivery.* — The engines and other apparatus called for under this contract shall be delivered and erection begun on the date when the engine room is ready to receive them, as determined by the engineer; and the contractor shall deliver all of the apparatus called for in this specification and have it ready for operation within two months after said date. The engine room will not be ready to receive the engines before ——.

*Dimensions.* — Each bidder must furnish with his proposal the following data and dimensions in full of each engine that he proposes to supply:

Dimensions of cylinder.
Minimum thickness cylinder walls.
Clearance of cylinder.
Width of piston face.
Diameter of steam and exhaust pipes.
Diameter of piston rod.
Length of connecting rod.
Diameter and length of main and outboard bearings.
Diameter and length of crosshead pin.
Area of crosshead shoes.
Diameter and length of crank pins.
Diameter and weight of flywheel.
Diameter of shaft in middle.
Weight of entire engine.
Guaranteed engine friction in per cent of rated load.
Guaranteed steam consumption at full normal generator load.
Guaranteed speed regulation in per cent of no-load speed.
Guaranteed momentary variation in speed under any change in load from no load to full load.

*Relation of Engine and Generator Builders.* — Within two weeks after the generator contract is let the generator builder shall send to the engineer in duplicate, and to the engine builder, shop drawings of the generator with the weights of the armatures. The engine builder will design the shafts, bearings, and foundations to accommodate the generators. The generator builder shall at the proper time furnish the engine builder with a gauge giving the exact diameter of the armature bore and a drawing showing location and size of the key-way. The engine contractor will then be responsible for furnishing a shaft and key of the size given by the gauge and drawings. The shaft must fit the armature in a manner satisfactory to the engineer. The generator contractor shall force the armature upon the engine shaft after delivery at the building. The engine contractor and generator contractor shall coöperate as directed by the engineer in erecting each engine and generator. The engine contractor and generator contractor shall each be individually responsible at all times for all parts of the apparatus which each is to furnish. The engine builder will furnish foundation bolts, inclosing pipes, and anchor plates for both engines and generators, and he shall construct the engine and generator foundations.

*Inspection.* — The engineer reserves the right to inspect the engines at any time during their construction and installation, and the contractor shall afford representatives of the engineer all facilities to this end.

*Drawings.* — Bidders shall submit with their proposals blue-prints in duplicate showing both in plan and elevation dimensioned drawings of the general arrangement and the principal dimensions of engines they propose to supply.

Within thirty days after the engine and generator contracts are let the engine builder shall furnish to the engineer four sets of blue-prints for the foundations for the engines and generators, together with detail drawings of

all foundations, bolts, anchor plates, nuts, washers, etc.; these will be required at the earliest possible moment for use in preparing the working drawings. Should the engineer ask for them the contractor shall, within thirty days after the contract is let, submit a complete set of shop drawings showing the engine frames, cylinders, valves, valve gear, piston, crosshead, flywheel, and bearing, and of such other parts as may be called for, which drawings shall be approved by the engineer and returned to the bidder before work is commenced. If the designs submitted are unsatisfactory to the engineer, they shall be changed to conform with the requirements of these specifications, and the engines shall be constructed in accordance with these requirements. It shall be distinctly understood that the approval of each or any drawings will not relieve the contractor from the requirements of the specification.

*Type.* — Engines shall be simple Corliss type engines as hereinafter specified, and adapted for the duty demanded in driving direct-current electric generators, the armatures of the engines being mounted on the engine shafts.

*Capacity.* — The normal rating of two of the generators will be 500 kw. at 100 r.p.m.

The normal rating of one of the generators will be 250 kw. at 100 r.p.m.

Each engine must be capable of driving continuously its respective generator at normal output and at normal speed with an initial steam pressure of 125 pounds above the pressure of the atmosphere at the throttle valve and a back pressure in the exhaust pipe outside of the cylinder of 2 pounds above the pressure of the atmosphere. The engines shall be sufficiently strong in all their parts to do this work, and the valve gear must be so designed to permit the generator overloads mentioned (50 per cent) to be maintained.

*Speed.* — The normal speed of the engines shall correspond to the normal load speeds of the generators given above.

*Steam Pressure.* — The steam pressure at the throttle valves of the engines will be 125 pounds. The engines must be sufficiently strong to stand a working pressure of 150 pounds should it be desirable to raise the steam pressure to that extent.

*Design.* — The engines must in every way conform to the best modern practice.

*Beds.* — The bed of the engine must be of heavy design, with a single casting containing and cast with the main pillow block and inclosing the crank pits. The girder type of frame will not be accepted. The bed shall have a solid bottom and be well stiffened with ribs. The crank pit shall be designed so as to catch and retain the oil, and all exposed flat surfaces adjacent to the crank pit shall have a gradual slope towards the crank pit for draining into the same. Guides shall be designed to drain into crank pit only. Means shall be provided for draining off the oil through piping as directed. The entire bed must be of massive design, with smooth surfaces and well-rounded corners, and must extend to the foundation at all points. The bed plate shall be anchored by a sufficient number of properly spaced foundation bolts seating on raised and counterfaced bosses.

*Guides.* — Guides shall be constructed to insure perfect rigidity and alignment and give access to working parts. They may be cast with bed plate, or separately and bolted to the latter. The guides must be accurately bored,

true to the axis of the cylinder. If guides are cast separate from the bed plates, they shall be bored and both ends faced off at one setting. If cast in one piece, guides shall be bored and cylinder ends faced off at one setting. Oil receptacles shall be provided at front end of the lower guide arranged to drain into the crank pit.

*Shafts.* — The engine shaft shall be of high-quality open-hearth steel hydraulically forged. The crank shaft shall be ample in size to carry the weight of the flywheel and armature, and of ample length to afford proper working space. Shafts shall be free from all defects, turned at the proper diameter, and polished when exposed. The engine shafts shall be suitable for the particular make of generator purchased. That part of each shaft supporting the flywheel and armature shall be of greater diameter than the bearings so that the radial distance from the center of the shaft to the bottom of the key-way shall be greater than the radius of the shaft at the bearing by one-half inch. Change in diameter shall be an easy taper of 15 degrees with axis of shaft, if space will permit. The ratio of the diameter of the journal to its length shall be not less than 1 to 1.6 and not more than 1 to 2.

*Cranks and Crank Pins.* — Cranks shall be of the counterbalanced disk type turned and polished on the face and rim. Crank disks shall be forced upon the shafts by hydraulic pressure and properly keyed. Crank pins shall be of open-hearth forged steel, shouldered and forced into crank disks by hydraulic pressure. Crank pins shall be of such a size that the effective pressure on the projected area of the pin shall not exceed 800 pounds per square inch.

*Cylinder Sizes.* — Bidders shall state in their proposals what cylinder sizes they propose to use. The use of a relatively large cylinder diameter and short stroke will not be permitted. Cylinders shall be of such a length between heads as not to require an unusually long piston.

*Valves.* — Engine valves shall be of the multiported Corliss type, so designed that they will not make a shoulder in the seats. Valves shall be machined and ground on a grinding machine to the proper diameter to insure tightness without undue friction. Valves must be so designed that they may be removed easily through the openings in the back of the cylinders without disturbing the valve gear. Valve setting marks shall be made on the back of valves. Valve stems shall be of steel accurately ground to gauge and operating in bronze bushings. Valve stems shall be provided with suitable packing, and in addition it is desired that flat steel disks carefully machined and ground be shrunk on the valve stems so that when they bear upon the valve bonnet the joint will be steam-tight.

*Cylinders.* — Cylinders shall have heavily ribbed walls of ample strength. The cylinder walls shall be sufficiently thick that when they are bored to one-half inch larger diameter the tensile strain in the metal will not exceed 2000 pounds per square inch with a steam pressure of 125 pounds in the engine cylinder. They shall be of close-grained air-furnace iron as hard as can be machined, free from defects of any kind. Cylinders to be so designed as to permit expansion to occur at rear end through suitable guides. The piston bore shall be straight, of uniform diameter from end to end, and its surface is to be smooth, even, and free from flaws and defects. Ends shall be slightly counterbored. Cylinder heads and pistons shall be faced to insure

a definite clearance at each end. Clearance shall be reduced to a minimum. Flat surfaces must be strongly ribbed. Cylinders shall be drilled and tapped for indicator cocks. Cylinder heads shall be machined all over their inside surfaces, and they shall be provided with a polished false cover. Cylinders shall be covered with 85-per-cent carbonate-of-magnesia nonconducting material and then lagged with heavy planished-steel lagging to extend down to floor. Corner strips of polished steel, fastened with machine screws, shall be provided.

*Valve Gear.* — The valve gear shall be of the Corliss type with dashpots. Dashpots shall be of the noiseless, self-contained type of approved design. The entire valve gear shall be noiseless in action. Steam and exhaust valves shall be operated by means of separate eccentrics, with gear so designed that the cut-off can occur as late as three-quarter stroke. Valve gear shall have an approved releasing clutch or hook rod for both steam and exhaust so they may be operated by hand. All parts of valve gear must be adjustable to take up wear without changing the length of the connections. All pins in the valve gear shall be forged steel hardened and ground to gauge. All link and rod ends in valve gear shall be of phosphor bronze with graduated wedge adjustments. The wedges shall have a full bearing surface on the boxes. End set-screw adjustments will not be accepted. All stub ends shall be screwed upon the rods and locked by means of case-hardened hexagonal sleeve nuts. Hook rods shall be of forged steel. All similar parts of the gear shall be interchangeable. The drip pipes for the valves shall be brass nickel-plated. They shall be carried below the floor as directed.

*Governor.* — Governors in design and operation shall be equal to the best practice. They shall be so constructed that the speed of the engines will not vary more than three revolutions per minute under any change in load from no load to full load applied suddenly or gradually, and shall not be so sensitive as to cause racing or hunting. Their action shall be controlled if necessary by a suitable oil dashpot. The bottom of the governor spindles, if of the vertical flyball type, shall rest in a case-hardened steel step bearing. Upper end shall be provided with an oil hole fitted with a one-pint sight-feed lubricator. Governor weights, if of the inertia type, shall oscillate upon roller or ball bearings. Governor pins shall be of hardened steel ground true to size.

*Eccentrics.* — Steam and exhaust valves on all cylinders shall be operated by separate eccentrics. Eccentric straps shall be split and lined with Babbitt metal hammered in and turned to a smooth finish. Straps shall be held together by two through bolts with lock nuts. Each eccentric shall be held by at least two set screws. Eccentric straps shall be fitted with deep oil trough. Eccentric rods shall be of forged steel polished and fitted with adjustable box end of bronze. Each rocker and shaft shall be fitted with one-pint sight-feed oil cup.

*Flywheel.* — Flywheels shall be of cast iron, heavy, sound, and perfectly balanced. The wheels shall be cast in two sections with joints planed to a proper fit. The periphery and both sides of the rim shall run true. Face of rims shall be provided with barring holes, and barring device of approved design shall be provided for each engine. Rim joints shall be made with

heavy steel links or arrowhead keepers set into the sides of the rim at each joint. Through bolts shall be used in the hub. Bolts and links shall be of open-hearth steel. Bolt holes shall be drilled out of solid metal or they shall be reamed out to size. The flywheels shall be of ample size and weight to give proper regulation and to properly control the parallel operation of the generators. The wheels shall be so constructed that under no condition will intense stress due to improper manufacture occur.

*Connecting Rod.* — Connecting rod shall be open-hearth steel of ample strength, accurately turned and polished. Crank and crosshead ends of rod shall be of solid type. Boxes at each end of connecting rods shall be provided with wedge adjustments controlled by screws for taking up wear in such a manner as to preserve constant the length between the crosshead and crank pins. Crank pins shall be lined with Babbitt metal hammered in and bored out, crosshead boxes to be of phosphor bronze.

*Crosshead and Pins.* — Crosshead shall be of cast steel, and it must be connected to the piston rod in an approved manner. The crosshead shoe must have a large wearing surface, must be lined with Babbitt metal, and must be provided with wedge adjustments for taking up wear by means of through bolts set up by lock nuts. Shoe area to be such that pressure will not be over 35 pounds per square inch. End screw adjustments and fastenings will not be accepted. The crosshead pin shall be of open-hearth steel with ground taper fits on front and rear side of crosshead held in position by steel jamb nut.

*Piston Rod.* — Piston rod of each engine must be of open-hearth forged steel ground to gauge. Rod must be connected to the piston and to crosshead to permit adjustment for equal clearances. Means shall be provided for preventing piston rod from turning in the crosshead.

*Packing.* — Stuffing box for each piston rod must be provided with an approved metallic packing so designed as to be capable of removal without removing the piston.

*Piston.* — Piston must be of approved design with cast-iron self-adjusting packing rings. Piston rods shall be forced into the pistons and be provided with a steel check collar or other equivalent means, doubly secured to absolutely prevent it from working loose. Junk ring to be so arranged that packing may be examined, removed, or replaced without removing the piston from the cylinder.

*Bearings.* — The main and outboard bearings shall be of pillow-block type adjustable in a horizontal direction. Bearings must be provided with Babbitt cast into dowel pockets, well hammered in, turned, and scraped. Both bearings shall be capable of being removed by raising shaft $\frac{1}{8}$ inch. Wear shall be taken up by means of wedges on both sides having full bearing surface upon the shells. Set-screw adjustment will not be accepted. The pedestals for the outboard bearings shall extend downward sufficiently below the floor line so that the finished floor can be built in around the pedestal and cover the foundation. A polished false cover shall be fitted over the outboard ends of shafts. Bearings shall have heavy caps provided with large handhole and holding-down bolts. Caps shall be tapped for eyebolts which shall be supplied for each engine. The end of outboard bearings adjacent to the genera-

tors shall have means for preventing the oil from working its way into the shafts. The bearings shall be lubricated by rings or chain extending into an oil reservoir in the pillow block. The resultant bearing pressure shall not under any condition of pressure exceed 180 pounds per square inch of projected area. Pressures shall be considered as the vertical component of the 125 pounds pressure on the crank pin plus the pressure due to the weight of the flywheel, shaft, eccentrics, crank disks, and armature. Contractor shall provide some approved method of oiling the main bearings of each engine.

*Cylinder Fittings.* — Each cylinder shall be provided with polished nickel-plated brass drip valves and piping run below the engine-room floor. Each cylinder of each engine shall be provided with indicator piping with three-way cock, all of brass nickel-plated.

*Oil and Water Drips.* — Proper means shall be provided to collect all dripping oil and water. All pockets in each engine where oil and water will collect and all points requiring the drainage of oil or water shall be provided with a drip system, the design of which shall be furnished and approved. Pipe and fittings above the floor shall be of polished brass (iron-pipe size) nickel-plated. The oil drips for each engine shall unite in a common oil main one inch in size which shall be run to a point under the floor. There shall be a separate drip system for dirty drips, to be run to point under floor where the steam fitter will connect. Each opening in the floor for all pipes shall be protected by a nickel-plated cast-brass collar. Contractor shall do all cutting for all pipes. All necessary nickel-plated brass oil or water drip piping shall be supplied.

*Cylinder Base Pans.* — Cast-iron base pans shall be provided under each cylinder and valve gear so as to collect all oil and water.

*Oil Pans.* — The contractor shall supply proper and approved pans of heavy polished brass as may be required for the valve gear, rocker gear, shaft, gear, guide barrels, and for other points as may be necessary.

*Splashers.* — The contractor shall provide for each engine oil guards or splashers which shall be made of heavy planished sheet steel with polished steel corner pieces. The splashers shall be so constructed that they will completely cover the opening opposite the crosshead and house the crank and connecting rod. Approved doors giving proper access to all parts shall be provided in such oil guards. The splashers shall be made with flanged sections properly stiffened with angle iron, and shall be so constructed as to be easily removable. The joints shall be so constructed that no oil will work through them. Whenever openings are allowed the edges shall be well strengthened and protected by planished steel strips. A similar arrangement shall entirely close the eccentric, and this must be provided with sliding covers. The bottom of eccentric hoods shall be furnished with a drain for the drip pipe, and the entire structure must be securely fastened to the bed.

To protect the generator from oil creeping along the shaft, thin disks shall be furnished rotating just inside the eccentric hood, so that oil may be thrown into the hood before it can go farther. The space between the disk and shaft shall be so packed that no oil can creep through underneath.

*Indicators.* — The contractor shall furnish two outside spring —— indicators of the latest pattern, the pair in neatly finished box with lock and key. Each

indicator shall be provided with two indicator cords and the followings springs: 40 pounds, 60 pounds, and 80 pounds, one extra drum spring, and a set of graduated scales, one for each spring, and 500 metallic-faced cards. The contractor shall also supply three approved reducing motions, one for the small engine and two for the large engines, attached to engine, which can be thrown in or out while engine is running. Each engine shall be so fitted that the indicator and reducing motion may be connected to them.

*Wrenches.* — The contractor shall supply a complete set of neatly finished case-hardened wrenches for all engines, also such eyebolts as may be needed. Wrenches and eyebolts shall be mounted on a neatly finished and varnished wrench board of black walnut with beveled edges. Boards shall be mounted on the wall of engine room where directed by the engineer. Wrenches shall not be used in erecting the engine.

*Oiling Devices.* — Each cylinder shall be provided with approved and positive means for oiling. Each engine shall be provided with adequate and proper means for lubricating all moving parts. The contractor shall place ample sight-feed oil cups wherever they may be required and as directed by the engineer. The crosshead pin shall be provided with neatly finished and approved wiping device. An approved oiling device shall be furnished for each crank pin with necessary standard. Wherever grease cups are required compression cups shall be provided. Oil cups shall be of glass body, arranged for gravity and for hand feed, and shall be of —— make. Grease cups shall be of —— heavy-pattern compression type. All parts requiring lubrication shall be provided with receptacles for catching oil when the oil may be supplied by the oil filtering and circulating system, described under "Central Oiling System." All oil cups, pipes, and fittings above the floor shall be brass nickel-plated.

*Central Oiling System.* — The contractor shall furnish and install, complete and ready for operation, an Automatic Gravity Oiling System, for supplying machinery oil to each point of lubrication on two 500 kw. and one 250 kw. Corliss engine generating units and two oil sinks. Also a gravity cylinder-oil system, for supplying cylinder oil to two oil sinks.

A plan showing the location of the generating units and oil sinks, also the approximate location of pumps, filter, tanks, etc., may be seen at the office of the consulting engineer. This arrangement is subject to such modification as may be found desirable, to conform to building conditions. Trenches will be furnished by the builder, and combination oiling system sight feeds on the engines will be furnished by the engine builder. Steam connection to pumps will also be provided under a separate contract. The central oiling system will include the following apparatus: —

One gravity oil filter.

Two oil storage reservoirs.

Two steam-driven oil pumps.

Two brass oil sinks.

Two drip pans or pumps and filter.

One combination receiver and pump governor.

All piping, valves, unions, fixtures, appliances, labor, and material necessary to make a complete installation.

*Oil Filter.* — Furnish and install one No. 18 multiple-type —— oil filter, having a daily filtering capacity of not less than 250 gallons. Body of filter to be constructed of heavy galvanized sheet steel and jacketed with Russian iron and polished steel angles. Filter to be equipped with automatic water overflow.

*Oil Storage Reservoirs.* — Furnish and install two 75-gallon oil storage reservoirs, to be supported in a horizontal position by not less than two brackets. They shall be constructed of not less than No. 10 B.W.G. galvanized iron and equipped complete with all necessary flanged pipe openings, oil gauges, automatic overflow, etc.

*Oil Pumps.* — Furnish and install two steam-driven duplex oil pumps, full brass fitted and equipped with valves suitable for handling oil. Each pump to be mounted on suitable angle-iron base.

*Oil Sinks.* — Furnish and install two —— oil sinks, of heavy polished brass with beaded edges and supported on polished brass brackets. Each sink shall be provided with two bibcocks, removable tray with strainer and drain connection.

*Drip Pans.* — Furnish and install polished-brass drip pans for pumps and filter turned up ¾ inch and wired. Each pan to have necessary drain connections for carrying away drips.

*Oil-feed Lines.* — Furnish and install a machinery-oil supply line extending from the oil storage reservoir through the trenches to each engine and to two oil sinks. All branch feed lines to be provided with a gate oil valve, located just above the engine-room floor. Another supply line shall be installed from the cylinder oil storage reservoir to the two oil sinks. All piping above the floor line, extending up to and on the engines and oil sinks, shall be drawn-steel tubing, fully nickel-plated and connected up with nickel-plated pipe fittings. All pipes, fittings, valves, etc., on the engines shall be supported with nickel-plated concealed screw hangers, holding the pipe away from the engine frames a distance of about 1 inch.

*Drain Pipe.* — Furnish and install a used-oil return drain line, extending from the main return on the four generating units and two oil sinks to the receiver and pump governor, also a line for returning any excess oil delivered to the reservoir.

*Pump Connections.* — Suction and discharge headers on pumps to be arranged with necessary connections, valves, etc., so that either pump may be used for delivering dirty oil from the receiver to the filter, or filtered and cylinder oil from the filter to the overhead storage reservoirs. A connection for delivering new oil from the barrels into the system shall be made to the suction header and extended as determined at the time of installation. All pipe around pumps to be brass nickel-plated and all connections to pumps, filters, tanks, etc., to be made with ground-joint brass unions. The ends of all pipe are to be properly reamed, joints being made perfectly oil-tight, and plugged tees and crosses shall be used in lieu of elbows and tees.

*Pump Foundation.* — Foundations for pumps and filter to be of selected red brick set in Portland cement mortar-faced with best-quality white American enamel brick with rounded corners, set in white mortar. Provide 4-inch rubbed bluestone cap for foundation.

*Oil Pumps.* — Furnish and connect on each of the three generating units one three-feed —— oil pump connecting with both ends of the cylinder and with the metallic piston-rod packing.

*Finish.* — The engine shall be finished in three styles, — polished surfaces, machined surfaces, and plain surfaces.

Polished surfaces shall show no tool marks and be highly polished. These surfaces shall include such surfaces as have been elsewhere specified as polished surfaces, also the entire valve gear, the cylinder covers, connecting rod, governor rods, valve bonnets, dashpot covers, and throttle-valve bonnet. All exposed machined surfaces shall be free from tool marks. Plain surfaces shall be filled and rubbed down as many times as necessary to secure a smooth and even finish and put them in proper condition for painting.

*Throttle Valves.* — The throttle valves of all engines shall be of extra-heavy-weight iron-body globe valves with removable bronze seats and bronze mountings, with bodies tapped for drip pipes. Valve seats shall be ground so as to be steam-tight. Valves shall be of the outside screw type.

*Painting.* — The engine builder shall paint and varnish all engines and generators. All parts of the engines not polished or covered with planished material must be thoroughly cleaned, primed, rubbed down, and given two coats of lead and linseed oil at the shop. After the engines are erected and ready for operation the engines and all iron castings in each generator shall be filled and rubbed down as many times as is necessary to give a smooth and even finish. The work shall be then given one coat of selected coach color, striped with gold leaf, which is to be followed by two coats of coach body varnish.

*Railing.* — The contractor shall supply and install for each engine a railing of 1½ inch pipe which shall be run around both sides of the flywheel and generator pit and terminate in an approved manner. All pipe stanchions and fittings and railing shall be of polished steel. The posts to be screwed into floor polished flanges.

*Foundations.* — The space available for engine foundations is shown on the accompanying plan, and it is proposed to use a continuous concrete block for the foundations, which the engine builder shall construct. If in the opinion of any bidder additional mass is necessary to prevent vibration, the foundations may be extended in a horizontal direction and the bidder shall figure upon such extension and notify the engineer in his proposal of what increase he deems necessary. If no such notice is given, it shall be assumed that the space provided by the engineer for foundations is satisfactory. The foundations shall be separated from the concrete below and from column footings and from all other building material by a bed of dry sand about 4 inches thick extending beneath and around all sides of the foundation.

Concrete to consist of one part Portland cement, three parts sand, and four parts broken stone that will pass a two-inch ring. Material to be mixed as directed and sufficient water added to make the mixture run lightly. Portland cement to be of approved make and to meet such tests as the engineer may prescribe. Sand to be clean and sharp and of approved quality. Concrete will be laid in layers of not over 6 inches thick and thoroughly rammed if so directed. Pockets for generator leads shall be provided as directed.

The top of foundation to be of such a height that the tile floor of the engine room can be laid over it and touch the base of the engine and generator. The generator pit shall be provided with curb angles with mitered corners properly secured into foundations so that tile may be laid flush with and against it. The generator pit shall be lined with white tile by the tile contractor. The engine contractor shall furnish substantial board templates for building the foundations.

When the engine and generator are set and blocked up on wedges, the contractor for the foundation will build a suitable dam around the edges of the engine bed and support for the generator, and he shall then supply and completely fill with grout the voids between the engine bed and the foundation and the space between the foundation bolts and inclosing pipes. Grout shall consist of one part Portland cement and one part sand with sufficient water to make a fluid substance. Particular care shall be taken before the grouting is commenced that the foundation bolts extend up through the foundation only by such a distance as will be required by a full nut.

Within thirty days after the engine and generator contracts are let the engine builder shall submit foundation drawings for approval, to be not less than scale of ½ inch to the foot. The engine builder shall be responsible for the accuracy of the drawings of the foundations for the engines and generators. He shall furnish anything needed at the proper time to prevent delay.

Each bidder shall furnish with his proposal the allowance he will make from his bed provided the owners construct the foundations and do the grouting and setting of curb angles.

*Foundation Bolts.* — A complete set of foundation bolts, machine-finished nuts, and all necessary anchor plates and pipes must be furnished. Anchor plates shall be so made as to prevent lower nut on foundation bolt from turning. A steel pipe of proper length and of a diameter about 2 inches larger than the diameter of the bolts shall be furnished with each foundation bolt.

*Bolts and Nuts.* — Case-hardened and finished polished nuts shall be used in all exposed work, and also for all parts requiring frequent removal and adjustment. All other nuts and bolt heads above the floor level shall be finished. All nuts in contact with polished surfaces shall be polished.

All nuts and bolt heads shall be hexagonal in shape and must be faced on top and bottom at right angles to the axis of the bolt. The sides of all nuts shall accurately fit corresponding wrenches. All nuts of pillow-block cap bolts and follower bolts of pistons, all screw joints in moving parts, and all keys shall be provided with a secure locking device. Through bolts shall be used in preference to studs wherever practicable.

*Workmanship.*—All fits shall be thoroughly machined. No shims will be permitted. No file fits will be allowed in the construction. Polished surfaces are to show no tool marks. All nuts on rough castings shall fit facings raised above the surface, except where otherwise directed. All flanges, collars, and offsets shall have well-rounded fillets. All unfinished surfaces shall be smooth, devoid of all imperfections, chipping, or other rough tool marks, and shall be properly cleaned, rubbed, etc.; at all joints the pieces joined shall be made symmetrical and shall be dressed so as to match properly and present a neat finished appearance. All bolted joints shall be ream-bolted. All rotating or reciprocating parts shall

be perfectly balanced when possible. All bright parts shall be carefully polished and then slushed before shipment with white lead and tallow. After erection they shall be thoroughly cleaned. No joints made by fracture will be permitted at any point. All valve rods, eccentric rods, stub ends, etc., shall have a polished finish. Throttle-valve spindles shall be polished all over and shall have polished handwheels. All valves must be capable of being packed while open and under pressure. All machine work shall be made according to the interchangeable system, and all surfaces which can be advantageously finished by wet emery grinding shall be so treated. No shrunk fits shall be used on any part. All pipes shall be full-weight standard. All fittings for steam pipes shall be extra-heavy flanged. All castings must have good-sized fillets at all reëntering corners, and the use of small brackets to stay flanges to the body of the casting will not be allowed. All flat surfaces and surfaces acted upon by pressure shall be substantially strengthened and stiffened with heavy ribs to make them of ample stiffness and strength to safely carry the loads to which they will be subjected.

*Attendant.* — A competent attendant must be furnished by the contractor to take charge of the engines at the time of the trial run. He shall remain in charge and be responsible for the engines until same have been accepted. He must be qualified and will be required to give to the chief engineer all necessary and proper directions regarding the operation of the engines and other apparatus supplied under this contract.

*Test.* — The engineer will make such proper tests as he deems necessary to find out if the engines and generators comply with the requirements of their respective specifications. Such tests will be made as soon as possible after the entire electrical and steam equipment is in a condition to test and a proper load can be obtained. The test will not be made, however, until the entire electrical equipment has been in regular and successful operation in a manner that is satisfactory to the engineer for at least one month, this period constituting the trial run referred to in the preceding article. During the trial run and tests the contractor for this work shall coöperate with the contractor for other parts of the work, and he shall give the engineer such assistance as he may require.

The friction load during the test shall be obtained by running the engine at its rated speed with a steam pressure of not less than 125 pounds, with the brushes of the generator not in contact with the commutator and the fields unexcited.

*Guarantees.* — The contractor shall guarantee to furnish engines that shall run so easily that no vibration will be noticeable in any part of the building; that shall operate perfectly, without abnormal wear, without heating, and without objectionable noise. The contractor shall guarantee to make good any defect in workmanship or material or defect of any kind that may occur within one year of the date of acceptance of his work.

# CHAPTER VI.

## TURBINES.

**Advantages of Turbines.**—The steam turbine has reached such a stage of development that in many cases it is superior to the steam engine, particularly in situations where high rotative speeds may be employed, such as the driving of alternating-current electrical generators, centrifugal pumps, blowers, and compressors. Turbines driving direct-current generators up to 700 kilowatts capacity that will give excellent satisfaction can be obtained, the earlier difficulties of commutation in the generator having been overcome. The particular points of advantage of the steam turbine are high efficiency under variation in load, simplicity, economy of space, the absence of oil in the exhaust steam, the ease with which alternators driven by steam turbines may be operated in parallel, and the slight falling off in efficiency due to use.

**Attention.** — There is no doubt but the turbine requires less attention than the reciprocating engine. There are few rubbing surfaces outside of the main bearings, and with these properly designed and lubricated the problem of looking after a turbine is a simple one.

**Space Conditions.** — As to the economy of space due to the use of a turbine in place of an engine, this feature holds true to a greater extent with large units than it does with small ones if the maximum economy of each type of prime mover is to be obtained. A reciprocating engine of an economical type will be more efficient than a steam turbine if moderate steam pressures and moderate vacua are employed, as with turbines the steam must expand through a high range of pressure if it is to work economically. Where from 26 to 27 inches of vacuum have been common with reciprocating engines a vacuum of from 28 to 28½ inches is desired for steam turbines, and this high vacuum can only be obtained by using more expensive and more elaborate condensing plants than would be required for the

lesser vacuum used with steam engines. Such condensing plants naturally take up a good deal of space, and where small turbines or even turbines of moderate size with such condensing plants are used it is doubtful if there is any saving of space due to the use of a turbine over that required by a steam engine with its condenser. There is one point in favor of the turbines, and that is they require less expensive foundations. One type of vertical turbine is now made for very large plants with the surface condenser forming the turbine base. It is interesting to note in this connection that during the year 1911 the New York Edison Company replaced the 3500-kw. units driven by three-cylinder compound engines installed in the Waterside Station No. 1 by vertical turbines of 20,000 kilowatts capacity each. The actual floor area occupied by the engine-driven unit was 918 square feet and by the turbine 297.5 square feet, so that the replacement will permit the development of six times the power in one-third of the space. It is interesting to note, in passing, that the steam consumption of the engine-driven units, while running at a speed of 73 r.p.m. with a steam pressure at the throttle of 185.6 pounds with a vacuum of 27.25 inches, while indicating 5442 horse-power, amounted to 11.93 pounds per horse-power per hour and 16.74 pounds of steam per kilowatt hour. The turbines replacing them are designed to operate with a steam pressure of 175 pounds and a vacuum of 28½ inches and with steam superheated 100 degrees. Under these conditions they are guaranteed to operate with a steam consumption of 15 pounds per kilowatt hour when developing 20,000 kilowatts, 14.4 pounds when developing 15,000 kilowatts, and 15.0 pounds when developing 10,000 kilowatts. Even when allowing for the more favorable conditions as to superheat and vacuum in the turbine, the turbine has somewhat the best of it in so far as economy is concerned.

**Absence of Oil.** — The absence of oil in exhaust steam from turbines makes their use advisable where exhaust steam free from oil may be used for some manufacturing purpose. It is also a decided advantage in eliminating the troubles due to oil in boiler-feed water where a surface condenser is used permitting the water to be used over and over again.

**Overload Capacity.** — The unusual overload capacity of a steam turbine, which is often referred to as being of particular

merit, depends entirely upon the rating of the turbine and of the generator to which it is connected, if it is an electrical generating unit. With a steam engine as ordinarily proportioned the point at which the engine speed begins to fall off to such an extent as to be troublesome is usually at overloads of from one-third to one-half of its rated capacity. There is no reason, however, why the cylinders cannot be made so as to withstand still greater overloads if the electric generator is built to stand it.

**Variable Loads.** — The steam consumption of a turbine, or at least some turbines, varies less at partial loads or at overload than it does with reciprocating engines, as will be seen by an examination of the data upon both types of prime movers under varying loads given in this volume.

**Durability.** — The indications are that a turbine, if properly constructed, is less affected by use, in so far as its steam consumption is concerned, than is the reciprocating engine, where the rubbing of the valves and piston will cause piston and valve leakage that will affect the economy to a varying extent depending upon the amount of wear. Wear does occur in buckets of some turbines, particularly when moisture is present in the steam. While it is believed that with superheated steam and properly made buckets the wear is negligible, it may be too soon to say that no wear occurs.

**Relative Operating Costs.** — Some interesting data given in Table 14 was furnished by Mr. Henry G. Scott, chief engineer of the Interborough Rapid Transit Company in New York City, an engineer of large experience in large power-house operations, in the Proc. Am. Inst. Elec. Engrs. in 1906, bearing upon the relative cost of operation with turbines and with reciprocating engines for street railway power houses of large size.

Table 15 shows the steam consumption per kilowatt hour that might be expected from electrical generators connected to various types of steam engines and to steam turbines.

Steam superheated 125° F.

The chances are that a well-designed Corliss engine driven unit of large size will operate quite as economically as a steam turbine of equal capacity when operating under conditions equally favorable to both.

TABLE 14. — RELATIVE COST OF OPERATION OF TURBINES AND ENGINES.

| Maintenance. | Reciprocating engine. | Steam turbine. |
|---|---|---|
| 1. Engine room, mechanical | 2.57 | 0.51 |
| 2. Boiler room | 4.61 | 4.30 |
| 3. Coal- and ash-handling apparatus | 0.58 | 0.54 |
| 4. Electrical apparatus | 1.12 | 1.12 |
| *Operation.* | | |
| 5. Coal- and ash-handling labor | 2.26 | 2.11 |
| 6. Removal of ashes | 1.05 | 0.94 |
| 7. Dock rental | 0.74 | 0.74 |
| 8. Boiler-room labor | 7.15 | 6.68 |
| 9. Boiler-room oil, waste, etc | 0.17 | 0.17 |
| 10. Coal | 61.30 | 57.30 |
| 11. Water | 7.14 | 0.71 |
| 12. Engine-room mechanical labor | 6.71 | 1.35 |
| 13. Lubrication | 1.77 | 0.35 |
| 14. Waste, etc | 0.30 | 0.30 |
| 15. Electrical labor | 2.52 | 2.52 |
| Relative cost of maintenance and operation | 100 | 79.64 |
| Relative investment, per cent | 100 | 82.50 |

TABLE 15. — PROBABLE STEAM CONSUMPTION OF ENGINES AND TURBINES.

| | Steam pressure, pounds by gauge. | Vacuum, inches. | Pounds of steam per kilowatt hour. |
|---|---|---|---|
| High-speed compound engine | 90 | 0 | 48 |
| 100-kw. turbine | 150 | 0 | 50 |
| High-speed compound engine | 150 | 0 | 36 |
| High-speed compound engine | 150 | 26 | 30 |
| Corliss simple engine | 100 | 0 | 39 |
| Corliss simple engine | 100 | 26 | 32 |
| Corliss compound engine | 150 | 0 | 30 |
| Corliss compound engine | 150 | 26 | 20 |
| 100-kw. turbine | 150 | 27 | 32 |
| 500-kw. turbine | 175 | 28 | 20 |
| 5000-kw. turbine | 175 | 28 | 16 |

**Effect of Vacuum and Steam Pressure.** — With turbines of about 300 kilowatts capacity with a steam pressure of 150 pounds by gauge, a steam consumption of about 50 pounds per kilowatt hour might be expected if run noncondensing with a back pressure equivalent to the pressure of the atmosphere. With a back pressure of about 5 pounds by gauge there would be an in-

crease of about 10 per cent in the steam consumption. With the same turbine when exhausting into a vacuum of 28 inches, the steam consumption would be about 25 pounds per kilowatt hour.

As to the effect of varying the vacuum, steam pressure, and superheat in turbine work with a steam pressure of 175 pounds by gauge, there is an increase in the energy available in the steam of about 5 per cent by increasing the vacuum from 28 to 28½ inches. The gain due to an increase in steam pressure amounts to about 1 per cent for every 10 pounds increase in pressure at an expense of about one-tenth of 1 per cent in the total heat in the steam for every 10 pounds increase.

The superheating of steam 100° F. with a steam pressure of 175 pounds by gauge and with a vacuum of 28 inches increases the total heat supplied to the steam by about 4½ per cent, with a corresponding reduction in the steam consumption of the turbine of about 8 per cent, resulting in a net gain of about 3½ per cent.

TABLE 16.—ECONOMY TESTS OF STEAM TURBINES.

| Turbine. | Load, kilowatt. | Steam pressure, pounds absolute. | Superheat. | Vacuum. | Steam, pound per kilowatt hour. | B.t.u. per kilowatt hour. |
|---|---|---|---|---|---|---|
| Rummelsburg, A. E. G. | 4,179.6 | 179 | 289.3 | 29.19 | 11.95 | 15,665 |
| Carville, Parsons | 5,164.1 | 207 | 125.2 | 29 | 13.15 | 16,122 |
| Chicago, Curtis | 10,816 | 190 | 147 | 29.47 | 12.9 | 16,206 |
| Berlin Moabit, A. E. G. | 3,150 | 185 | 225 | 28.5 | 13 | 16,352 |
| Boston, Curtis | 5,195 | 179.5 | 142 | 28.8 | 13.52 | 16,578 |
| City Electric, San Francisco, Westinghouse | 8,563 | 183 | 59 | 28.18 | 14.427 | 16,850 |
| N. Y. E., Westinghouse | 9,830 | 192.2 | 96 | 27.31 | 15.15 | 17,778 |
| Brown-Boveri, Rhenish Westfalen | 5,128 | 176 | 193.7 | 28.47 | 14.32 | 17,790 |
| N. Y. E., Curtis | 8,880 | 192.5 | 108.5 | 28.1 | 15.05 | 17,940 |
| Pacific Curtis, Oakland | 8,775 | 194 | 72.95 | 28.03 | 15.95 | 18,735 |

Some interesting data upon the performance of turbines of large size were given by Mr. George A. Orrok in the Transactions of the American Society of Mechanical Engineers for 1910. They are contained in Table 16. The different turbines are operating under different conditions as to pressure, superheat, and vacuum, and therefore cannot be compared one with the other without taking these factors into consideration.

In a test of a 10,000-kw. normal-rating Westinghouse turbine by Mr. S. L. Naphtaly, in March, 1910, at the City Electric Company in San Francisco, the results given in Table 17 were obtained:

TABLE 17. — TEST OF WESTINGHOUSE TURBINE IN SAN FRANCISCO.

| Load in kilowatts. | Steam pressure. | Superheat, degrees Fahrenheit. | Vacuum, 30-inches par. | Water per kilowatt hour. | Steam per kilowatt hour.* |
|---|---|---|---|---|---|
| 5333 | 173 | 54 | 28.34 | 15.65 | 15.21 |
| 7972 | 171 | 58 | 28.28 | 14.58 | 14.11 |
| 8198 | 169 | 60 | 28.10 | 14.59 | 14.04 |
| 9173 | 167 | 59 | 27.90 | 14.57 | 13.88 |

* Corrected to 175 pounds steam pressure, 100 degrees of superheat, and 28 inches of vacuum.

Some additional data upon the steam consumption of small turbines at various loads will be found in the specifications for turbines.

**Exhaust-steam Turbines.** — By connecting a turbine to the exhaust of a noncondensing engine and supplying the turbine with a condenser so that it may operate condensing, the power that may be obtained from the same amount of steam may be increased very materially. While this may also be accomplished by attaching a condenser direct to the steam engine, there is the probability of obtaining greater capacity by expanding the steam through the lower stages in a turbine than in a steam engine, for the reason that cylinder condensation limits the number of expansions that it is advisable to use in a steam engine to about 36, while the number of expansions that is possible to obtain with a turbine is limited only by the cost of the condensing equipment necessary to obtain them.

**Specifications.** The following has been slightly condensed from a specification for small turbine generating sets for use on ship-board by the United States Navy Department.

Each set to consist of an electric generator driven by a steam turbine, both mounted upon a common bed plate, or having a common frame provided with ample supporting feet. To be complete with throttle valve, and for sets mounted upon a common bed plate to be fitted with bosses for hand-rail stanchions.

The set as a whole shall be as compact and light as is consistent with due regard to strength, durability, and efficiency. The maximum allowable normal

speed, weight, over-all dimensions, and end clearance for disassembling to be as shown in Table 18.

TABLE 18.

| Kilowatt. | Revolutions per minute. | Length over all. | Base width. | Height over all. | Width over pipe connections. | Clearance for assembly. | | | Weight in pounds. |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Top. | Turbine end. | Commutator end. | |
| | | Inches. | Inches. | Inches. | Inches. | Inches. | Inches. | Inches. | |
| 5 | 5000 | 52 | $18\frac{1}{4}$ | 28 | 25 | ...... | 3 | ...... | 950 |
| 10 | 4500 | 60 | 21 | 30 | 30 | ...... | 5 | ...... | 1,650 |
| 25 | 3600 | 90 | 32 | 40 | 40 | 5 | 6 | 5 | 4,300 |
| 50 | 3300 | 100 | 39 | 51 | $47\frac{1}{2}$ | 10 | 6 | ...... | 9,500 |
| 100 | 2400 | 130 | 62 | 82 | 72 | 10 | 6 | ...... | 17,500 |
| 200 | 1700 | 174 | 76 | 92 | 100 | 13 | 6 | 3 | 29,000 |
| 300 | 1500 | 175 | 76 | 92 | 100 | 13 | 6 | 3 | 31,000 |

The design shall provide for accessibility to all parts requiring inspection during operation, or adjustment when under repair. Sets of 100-kw. capacity and larger shall be provided with coupling between the armature and turbine wheels to permit the removal of the armature without disassembling the turbine. Only sets with turbine directly connected to the generator will be approved.

The set must be capable of running without undue noise, excessive wear or heating; must be balanced and run true at all loads, up to $33\frac{1}{3}$ per cent above rating; and must be capable of running for long periods under full load.

Cast or wrought iron shall not be used for bearing surfaces. Both upper and lower halves of main bearings are to be removable without removal or displacement of shaft, on sizes of 50 kilowatts and over.

Suitable thrust bearings will be provided to prevent such movement of the shaft in direction of its length as might be caused by pitching of the ship with set erected with its shaft extending fore and aft.

The bed plates, where used, to be a substantial casting provided with accurately spaced, drilled holes for securing to foundation, and with lugs for securing hand-rail stanchions.

The seats for all bolt heads and nuts to be faced. All external nuts subject to frequent use to be case-hardened, and all nuts to be United States standard size. Where liable to work loose from vibration, nuts to be securely locked. All bolt ends to be neatly finished.

Adjoining portions of the machinery, where efficiency and good operation are affected by alignment, shall be doweled or rabbeted together, dowels to be fitted to accurately reamed holes. Through dowels in all cases to be provided with threads and a nut for withdrawal. Adjoining portions of the machinery shall be given corresponding marks, when desirable for insuring correct assembly.

Necessary wrenches, lifting eyes, jacking-off bolts for removing wheels, and all special tools, feelers, etc., required for assembling and disassembling the set to be furnished, mounted on a board, suitable for mounting on bulkhead.

Interchangeability among the different sets and their spare parts, of the

same size and make, as furnished in any one contract, is required. This is to be demonstrated as part of the final test for acceptance.

The general appearance of the set resulting from design and workmanship must be of the highest character. Any defect not caused by misuse or neglect, which may develop within the first six months of service, to be made good, by and at the expense of the contractor.

The works in which the construction of the machines is being carried on shall be open at all times during working hours to the inspecting officer and his assistants. Every facility shall be given such inspectors for the proper execution of their work.

The contractor shall make, at his own expense and previous to delivery, sufficient tests to insure that the sets conform in all respects to the specifications.

To prevent delays and additional cost to the Government of repeated tests, no more than two tests will be made after delivery of the set, the second test to be made within such time after the first test as may be stipulated by the Government. Failure to make the necessary repairs or remedy defects within that time will be cause for final rejection of the set. If left upon Government property while undergoing repairs, the risk of loss by fire will be the contractors.

It is the intention of these specifications to produce a generating set which shall be first-class in every particular relating to design, construction, or operation. Any omission from these specifications of any part of the machinery necessary to produce above results, or failure to describe design and construction of same, shall not operate to release the contractor from supplying such part of the machinery or from performing such work as part of the contract, and without additional expense to the Government.

The contractor shall forever protect and defend the United States in the full and free use and enjoyment of any and all rights to any invention or device which may be used in the construction, or whose use may be required for the successful operation of the generator sets, against any and all persons or corporations.

### Turbine.

The turbine shall be of the horizontal type, designed to ordinarily run condensing, but capable of operating at full load noncondensing with 5 pounds back pressure. The turbine to be of sufficient power when running condensing to drive the generator for an indefinite period at rated speed when the generator is carrying $1\frac{1}{3}$ load.

With normal steam pressure of 200 pounds, dry saturated steam, the steam consumption for $\frac{1}{2}$, $\frac{3}{4}$, 1, and $1\frac{1}{3}$ loads shall not exceed the amounts given in Table 19.

Superheating may be employed to insure dry steam, a proper correction being applied to the water rate to compensate for the degree used.

On all turbines operating condensing, the valve gear shall work automatically over the total range of conditions without adjustment of hand valves or similar devices. The words "total range of conditions" shall be interpreted to mean operation from zero to full load condensing, when operating with normal steam pressure and minimum vacuum of 23 inches. For overload and noncondensing operation, sets may be equipped with one hand valve.

TABLE 19. — WATER CONSUMPTION.

| Rated kilowatt. | Kilowatt load. | Vacuum. | | | |
|---|---|---|---|---|---|
| | | 28 inches. | 27 inches. | 26 inches. | 25 inches. |
| 5 | 3 | 87.75 | 88.25 | 89.00 | 89.50 |
| | 4.5 | 68.25 | 68.50 | 69.00 | 69.25 |
| | 6 | 56.00 | 56.50 | 56.50 | 57.00 |
| | 8 | 50.50 | 50.75 | 51.00 | 51.50 |
| 10 | 6 | 60.50 | 64.75 | 68.50 | 73.00 |
| | 9 | 49.25 | 52.00 | 54.50 | 58.00 |
| | 12 | 44.25 | 46.25 | 48.50 | 52.00 |
| | 16 | 41.25 | 42.25 | 43.25 | 45.00 |
| 25 | 15 | 54.75 | 55.50 | 56.25 | 57.50 |
| | 22.5 | 45.75 | 46.50 | 47.00 | 48.50 |
| | 30 | 40.00 | 40.25 | 40.50 | 42.00 |
| | 40 | 38.25 | 39.00 | 39.50 | 41.00 |
| 50 | 30 | 37.75 | 40.00 | 42.50 | 45.00 |
| | 45 | 32.00 | 34.00 | 35.50 | 37.00 |
| | 60 | 29.25 | 31.00 | 32.75 | 34.50 |
| | 80 | 26.75 | 28.50 | 30.00 | 31.50 |
| 100 | 50 | 36.00 | 38.75 | 41.50 | 44.00 |
| | 75 | 31.00 | 33.50 | 35.50 | 38.00 |
| | 100 | 28.00 | 29.50 | 31.00 | 33.00 |
| | 133.3 | 28.00 | 29.50 | 30.75 | 32.00 |
| 200 | 100 | 36.25 | 39.00 | 41.50 | 43.00 |
| | 150 | 30.50 | 32.50 | 35.00 | 36.00 |
| | 200 | 26.50 | 28.00 | 29.75 | 32.00 |
| | 266.6 | 26.50 | 28.00 | 29.50 | 31.00 |
| 300 | 150 | 31.00 | 33.25 | 35.50 | 36.50 |
| | 225 | 27.50 | 29.25 | 31.00 | 32.00 |
| | 300 | 25.50 | 26.75 | 28.00 | 29.50 |
| | 400 | 26.50 | 27.00 | 28.50 | 30.00 |

Buckets and reversing vanes shall be of material which will not injuriously rust or corrode under the action of steam.

The turbine to run smoothly and furnish the required power for full load at any steam pressure within 20 per cent (above or below) of those given in the table, and exhausting to condenser at 25 inches of vacuum; to furnish power for 90 per cent of full load at steam pressure 20 per cent below normal, and for full load at any steam pressure between normal and 20 per cent above normal, when exhausting into the atmosphere. It shall bear without injury the sudden throwing on or off of one and one-third times the rated load of the generator by making and breaking the generator's external circuit.

Turbine to have steam flanges on the right or left side, and exhaust flanges at bottom pointing down, or at bottom pointing to right or left, as may be

specified for the different installations. All piping shall be supported at points close to the turbine so as not to affect the alignment of parts or cause undue strain in turbine casing.

The steam inlet to be fitted with throttle and emergency valves (combined in one valve, if preferred) equipped with strainer intervening between the valves and the steam line. The emergency valve will be connected to the emergency governor in such a way that it will automatically close between 7 per cent and 10 per cent above normal speed of the turbine. Valve flange drilling to conform to specifications of the Bureau of Steam Engineering.

The governor shall be of the centrifugal or inertia type, designed to regulate the speed within specified limits.

Lagging shall be fitted as extensively as practicable to the turbine. It shall be done after preliminary acceptance of the turbine, in order that any defects in casting or joints may be readily found. The arrangement for securing the lagging in place shall admit of its ready removal, repair, and replacement.

The speed variation will not exceed 2½ per cent, when load is varied between full load to 20 per cent of full load gradually, or in one stop, turbine running with normal steam pressure and vacuum. A variation of not more than 3½ per cent will be allowed when full load is suddenly thrown on or off the generator with steam pressure constant between normal and 20 per cent above normal, a variation of not more than 3½ per cent when 90 per cent of full load is suddenly thrown on or off the generator with constant steam pressure at 20 per cent below normal, exhausting in both cases either into a condenser or the atmosphere. No adjustment of the governor or throttle valve during the tests shall be necessary to insure proper performance under the above conditions.

The turbine will operate without the use of lubricants in the steam spaces. Forced lubrication will be used on all bearings on which the shaft pressure exceeds 20 pounds per square inch of projected bearing surface. In case of forced lubrication, bearings to be cooled by water circulating in coils. These coils to be connected to the water system outside the pedestal.

Mandrels, with collars, complete, will be furnished for renewing the white metal of all bearings so fitted.

The material and design of the turbine will be such as to safely withstand all strains induced by operation at the maximum steam pressure specified, and by tests noted below.

The following shall be provided with each set as indicated:

(*a*) One 6-inch brass combination gauge for exhaust casing, graduated 30 inches vacuum and 30 pounds pressure.

(*b*) Two 4½-inch steam seal gauges, 30 pounds scale.

(*c*) Where forced lubrication is used, one 4½-inch oil pressure gauge to be furnished at the pump, and on sets of 200 kilowatts and above; a gauge also to be furnished for each bearing.

(*d*) An automatic device shall be installed on the turbine which shall shut off the steam by closing the throttle or emergency valve when the back pressure in the exhaust casing reaches between 10 and 15 pounds per square inch. The device to be so designed as not to be liable to impair the vacuum, and

shall not be of a piston or other type, that may stick, due to infrequent use.

(*e*) A pop safety valve not larger than $\frac{3}{4}$ I.P.S. exhausting to the atmosphere, shall be fitted to the turbine, of size sufficient to prevent injurious back pressure, due to possible leakage of the throttle valve.

(*f*) The assembled turbine shall stand a hydrostatic test of 50 pounds per square inch.

(*g*) Sets of 50-kw. capacity and above to be provided with a permanently connected direct-reading tachometer. All sets to be so designed as to permit the use of a portable tachometer or counter to determine speed.

(*h*) Where forced lubrication is used, provision to be made on the discharge side to permit the observation of the flow of oil and cooling water from each bearing. An adjustable relief valve to be supplied as part of the oil pump, which shall be of the rotary type.

(*i*) Oil tank, when employed, to be supplied with sight gauge and a $\frac{3}{4}$-inch bibcock at lowest point to permit withdrawal of water from the bottom of the tank.

(*j*) Provision shall be made for mounting commutator truing device, one of which shall be supplied for generators.

GENERATOR.

Generator to be of the direct-current compound-wound type, designed to run at constant speed and to furnish a pressure of 125 volts at the terminals, at rated speed with load varying between no load and one and one-third times rated load.

The magnet frame to be circular in form with inwardly projecting pole pieces. Sets of 50 kilowatts and above are to have the magnet frame divided horizontally, the two halves being secured with bolts to allow the upper half with its pole pieces and coils to be fitted to provide for inspection or removal of armature. The pole pieces shall be bolted to the frame. The magnet frame of sets with bed plate to be provided with feet of ample size to secure a firm footing.

Facilities to be provided for vertical adjustment of the magnet frame of sets with bed plate.

The laminations for the armature will be accurately punched from the best quality of thoroughly annealed electric sheet steel; slots to be punched in the periphery of the laminations to receive armature windings. The laminations will be insulated from each other and will be assembled on the spider or shaft and securely keyed. Laminations will be set up under pressure and held securely by end flanges.

The commutator bars will be supported on the shaft so that no relative motion can take place between the windings and bars. The bars will be of hard drawn copper finished accurately to gauge. The insulation between bars will be of carefully selected mica of not less than 0.025 inch thick. The bars will line with the shaft and run true and will be securely held in place by means of clamping or shrink rings.

The brushes will be composed of carbon. Each brush will be separately removable and adjustable without interfering with any of the others. The

point of contact on the commutator will not shift by the wearing away of the brush.

Brush holders to be staggered in order to even the wear over entire surface of commutator; the generator to be provided with provision which will permit shifting all the holders simultaneously. All insulating washers and bushings to be damp-proof and unaffected by temperature up to 100° C.

Finished armature to be true and balanced both electrically and mechanically, and run smoothly and without vibration. The shaft to be provided with suitable means to prevent oil from bearings working along to armature.

All copper wire to have a conductivity of not less than 98 per cent.

The main field coils, and commutating coils, if any, respectively, of any one set to be identical in construction and dimensions, and to be readily removable from the pole pieces. The shunt coils as well as the series coils are to be connected in series.

Connection boards will be mounted on the generator with necessary terminals for the main, equalizer, and shunt cables. Sets of 30 kilowatts and above to be provided with pilot lamp on the machine.

The field rheostat to be of fireproof construction, suitable for mounting behind the switchboard unless otherwise specified; to be provided with a shaft projecting through to the front, either directly connected or by sprocket chain. Hand wheel for front of switchboard to be supplied and to be marked to indicate direction of rotation for raising and for lowering the voltage of generator. The total range of adjustment to be from 10 per cent above to 20 per cent below rated voltage. Variation to be not more than ½ volt per step at both full and half load.

### Operation of Generator.

The compounding to be such that with turbine working within specified limits, field rheostats and brushes in a fixed position, and starting with normal voltage at no load or at full load, if the current be varied step by step from no load to full load, or from full load to no load, and back again, the difference between maximum observed voltage and minimum observed voltage shall not exceed 2½ volts.

The compounding and heat run (full load and overload) of the generating sets must be made with identical brush positions.

The dielectric strength for resistance to rupture shall be determined by a continued application of alternating E.M.F. of 1500 volts for one minute. Test of dielectric strength shall be made with the completely assembled apparatus and not with the individual parts, and the voltage shall be applied between the electric circuits and surrounding conducting material.

Insulation resistance shall be not less than one megohm.

With brushes in a fixed position, there shall be no sparking when load is gradually increased or decreased between no load and full load; no appreciable sparking when load is varied up to one and one-third times rated load; no detrimental sparking when one and one-third load is removed or applied in one stage.

The temperature rise of the set after running continuously under full load for four hours must not exceed the following:

| | Deg. C. |
|---|---|
| Armature, by thermometer | 40 |
| Commutator, thermometer | 45 |
| Series field coils, by thermometer | 40 |
| Shunt field coils, by resistance method | 40 |
| Shunt rheostat, exposed surfaces, by thermometer | 75 |
| Series shunt, by thermometer | 40 |

The jump in voltage must not exceed 15 per cent when full load is suddenly thrown on and off.

The rise in temperature to be referred to a standard room temperature of 25° C. Room temperatures to be measured by thermometers placed three feet from the generator so as to indicate the actual temperature of the room, but in no case to be within three feet of the turbine or other source of heat.

The generator to be capable of satisfactory operation for a period of two hours, carrying one and one-third times its rated full load in a room temperature of 35° C., under which condition the rise in temperature of no part shall exceed 60° C.

With two or more generators of the same manufacture operating in parallel, each machine shall not vary above or below its proportion of load ($\frac{1}{4}$ to $1\frac{1}{3}$ load) more than 12 per cent of its full-load normal current (provided the resistance of the line cables on the equalizer side of the machine terminals to the bus be such as to give within 10 per cent of equal drop in the cables of all machines, when carrying their respective normal currents). This distribution of current is illustrated as follows: With any number of machines in multiple, each of 1000 amperes rated capacity, the difference between the average load upon all machines and the load carried by any one machine shall not exceed 120 amperes or 12 per cent of the normal rating of the machine.

# CHAPTER VII.

## ARRANGEMENT OF STEAM AND WATER PIPING.

**Drawings.** — After contracts have been let for the engines, boilers, pumps, feed-water heater, and other auxiliaries of a plant, the engineer should obtain accurate drawings or blue-prints of each, showing in plan and elevation the exact location of all steam and water inlets and outlets. When the engines, boilers, and auxiliaries are located finally upon the plans, accurate drawings of the piping to connect them should be made. The piping should be shown in plan and in at least one elevation. The drawings should be to a scale of at least ¼ inch to the foot, and should show the location of every fitting and valve in the system. It saves time to indicate a valve by drawing correctly the position of its flanges and joining them by two crossed lines. When a number of fittings are to be placed close together, however, they should be drawn accurately and in full, for if this is not done it may happen that the piping cannot be put together on account of too much being crowded into the space allotted. An accurate drawing would prevent an error of this kind from occurring. Complete drawings result in lower bids, and do away with extras that are usually the result of incomplete or inaccurate drawings.

**Principles Involved.** — The fundamental object to be accomplished in steam piping is, of course, to carry steam without excessive loss of pressure; and next in importance is the requirement of safety, that the condensation loss shall be a minimum, and that the piping shall not leak. The greatest enemy to safety is the liability of water entering the system, or collecting in it, due to condensation; and it is, therefore, of particular importance that piping should be so constructed that it does not contain pockets in which water can collect. Pipes are usually proportioned so that steam travels at the rate of about one mile a minute, hence if a "slug" of water, as a body of water is sometimes called, is picked up by the steam and carried along with it, an accident is very apt to occur, either by the rupture of an elbow, at a change in direc-

tion of the pipe, or by the water entering the engine cylinder and wrecking it. In some instances pockets cannot be entirely done away with, but where they do occur they should be properly drained. Straightway globe valves, except when the stems are in a horizontal position, should not be used in a steam pipe, as the valve seat acts like a dam in forming a water pocket. Valves of any kind should never be placed so as to form a water pocket whether they are closed or open, if it is possible to locate them any other way. For instance, one frequently sees the stop valve on a boiler placed immediately above the boiler nozzle and a vertical section of pipe above the valve leading to an elbow from which a horizontal pipe leads to a steam main. When a boiler so connected is out of service, water due to condensation accumulates in the vertical pipe over the stop valve, and, although the pipe may be provided with a small drain pipe and valve, experience has shown that the latter are not always made use of to draw off the water, so that an accident may occur. It is just as effective to use an angle stop valve at the top of the vertical pipe mentioned and to pitch the horizontal pipe that leads from it to the main so that condensation will flow toward the latter. With such an arrangement water cannot collect, and hence an opportunity for an accident does not exist.

Steam pipes should always be pitched so that the condensation that occurs in them will tend to flow in the direction in which the steam is moving, the reason for this being that if it is attempted to run condensation against the current of steam, water hammer is quite likely to occur, as the water accumulates into a slug, which is finally picked up by the steam and carried along until projected against a fitting. If one wants to carry steam a long distance, from one point to another at a higher level, the pipe should be laid at the proper inclination, and at frequent intervals, say every 100 or 150 feet, the line should drop a few feet into a pocket that can be drained through a steam trap into a small return pipe running back to the boilers. A horizontal pipe should be inclined so that the condensation will tend to flow against the current of steam only when the pipe is excessively large, so that the velocity of steam flow will be much below the usual practice.

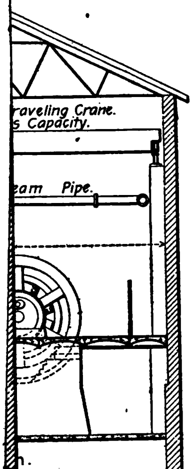
In systems of steam piping connecting several engines and the boilers supplying them, it is usually the custom to connect each boiler with a steam main from which pipes lead to the engines.

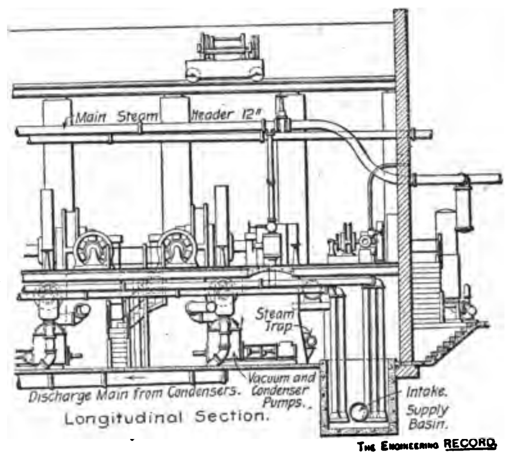
It is well to have this main of quite large size, so that the velocity of steam passing through it will be slow enough to allow any water that might be carried over from the boilers to accumulate in the main. The main, of course, should be drained by a pipe or pipes leading to a trap. If the main is divided into sections by valves, provision should be made for draining each section, for the reason that some parts may be shut off at times. Any branch to an engine should be connected to the top of the main to prevent, as far as possible, water from entering the branch.

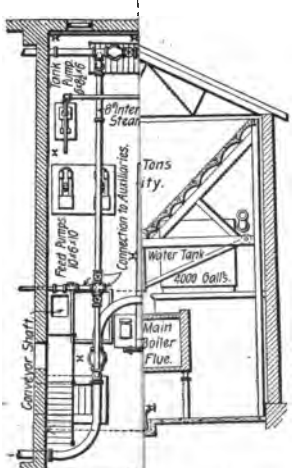
It is impossible to give definite information as to the design of piping for all steam plants, for the conditions met with vary so much. With electric power-house work, however, this is not the case, for the reason that the construction of these plants, unless they be very large ones, almost invariably follows one of two types that are standard as far as the relative location of engines and boilers is concerned. These types are as follows: that in which the boilers and engines are placed back to back, with a dividing wall between; and that with the boiler and engine rooms end to end, with the engines and boilers lying in the same direction. As far as the piping is concerned, the back-to-back type is the one most to be preferred, on account of the short and direct connection between the engines and boilers and the ease with which it can be enlarged. With the engine and boiler rooms placed end to end the condensation losses in the steam piping are greater, and this type of station, as far as the piping is concerned, is not easily enlarged. Very large stations are usually arranged in units, with an engine or turbine placed opposite to and connected with as many boilers as are required to supply it. Cross connecting pipes are used to connect the several units.

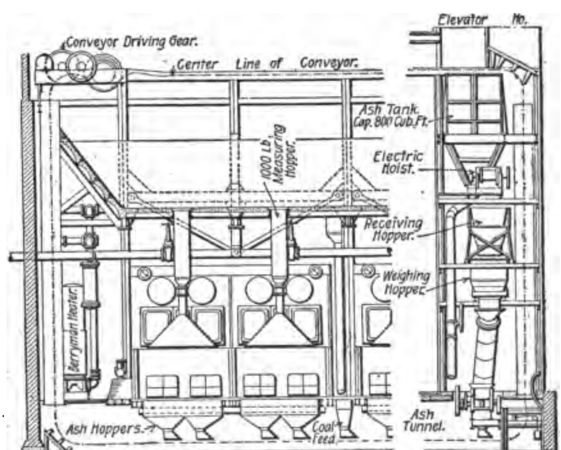
The steam piping for that type of station in which the engines and boilers are placed back to back and separated by a wall usually consists of a feeder from each boiler, connected with a main which is supported by the boiler-room wall or suspended from the roof trusses and also connected to each engine.

The engine-room floor is usually a little higher than that of the boiler room. The main and as much of the piping as possible should be located in the boiler room, for the reason that if an explosion occurred in some section of the pipe in the boiler room, it would be possible after the steam pressure fell to cut out the damaged section and operate the rest of the plant. If the engine




Longitudinal Section.

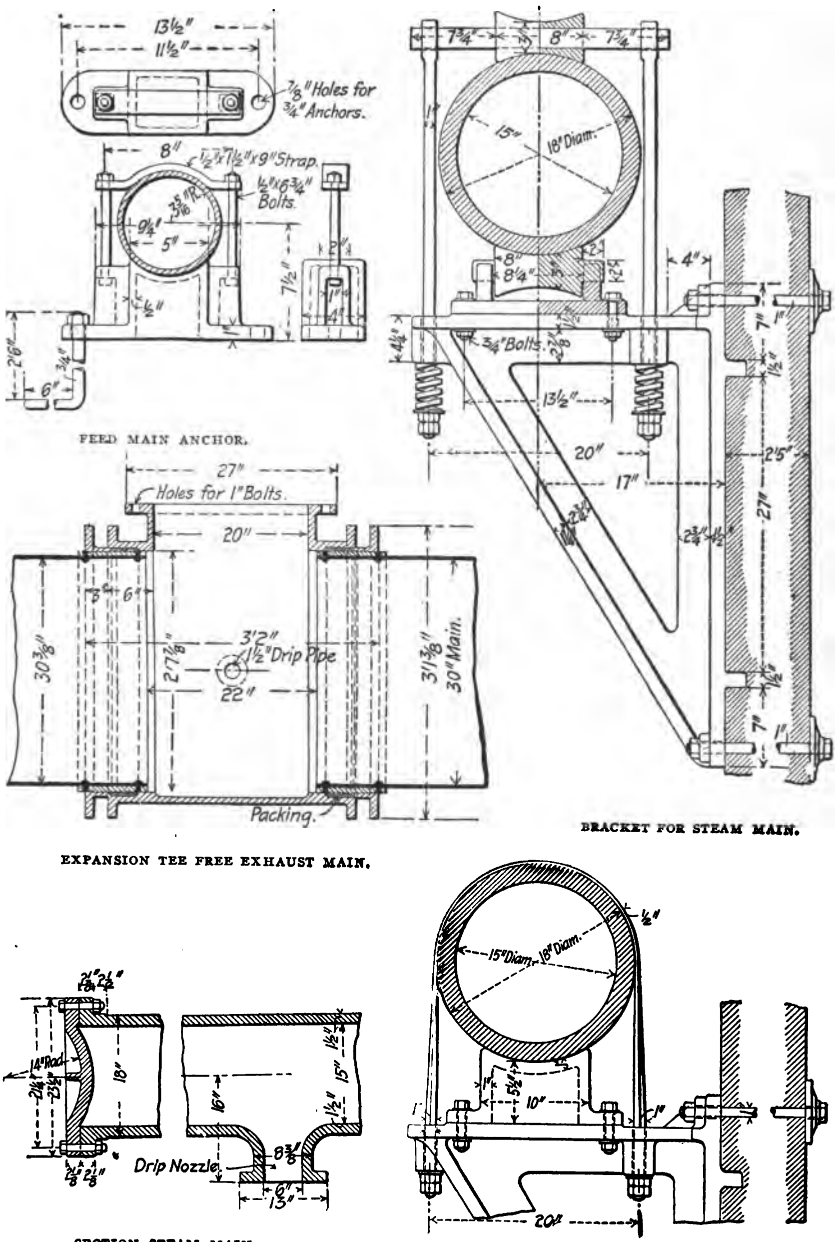



Fig. 23. Details of Steam Piping, Anderson Station (see Plate 6). (Sargeant and Lundy, Engineers.)

room, on the other hand, was the scene of the explosion and became filled with steam, the electrical apparatus would, in all probability, not be fit for service without considerable overhauling.

Two valves in a pipe leading from a boiler to a main are much more to be preferred than one, and are usually provided in large work. One should be close to the boiler, the other at the main. If there is only one, however, and the steam main is supported on a bracket on the boiler-room wall, it is a good arrangement to carry a pipe with a bend of long radius from the nozzle of each boiler to an angle stop valve bolted to a nozzle on the top of the main, as shown in Fig. 24, and to run a connection to each engine from a stop valve similarly located. Instead of the angle valve shown in the figure, an elbow with a gate valve adjacent to it, with the axis of the valve in a horizontal position, could be used. The valve in the pipe leading from the boiler to the main should not be placed over the boiler nozzle and thus form a pocket, for reasons previously explained. With both valves close to the main, they are easily reached from a light walk suspended from the roof trusses. Another advantage of placing the valve close to the main is that the condensation is less when the valve is closed. Some engineers prefer to place the angle stop valve immediately over the boiler nozzle and run a horizontal pipe from it to an elbow turning downward to a nozzle on the top of the main. The stop valve can then be reached by a person standing on top of the boiler setting. The engine connection should also be a bend of long radius. These connecting pipes will then have sufficient elasticity to permit expansion and contraction to occur without injury to the pipe. With the arrangement suggested, there is no chance of water collecting in any part of the system, save in the main, and this should be large enough so that the water can settle there and pass out by the drain-pipes. Frequently the pipe leading to each engine is provided with a separator close to the throttle valve. Besides intercepting moisture in the steam, the separator performs another function of great value, in that it provides a reservoir of steam close to the cylinder, which insures a higher and more uniform pressure in the cylinder up to the point of cut-off than there would be if it were omitted; and it also reduces the vibrations in the steam piping due to the intermittent flow of steam as the valves of the

engine open and close. For these reasons a separator, particularly if of large volume, is of much value.

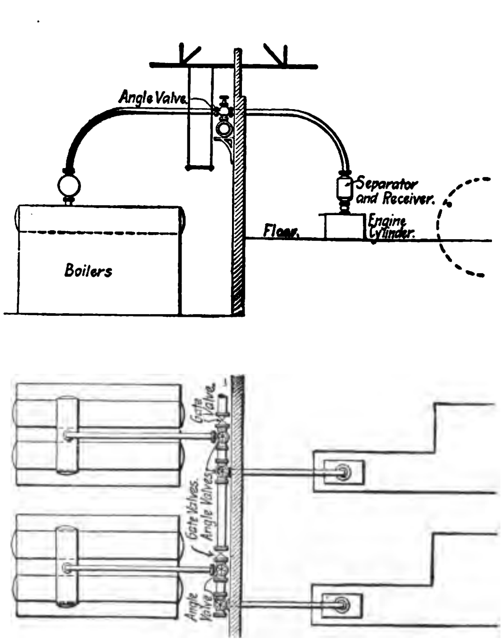



FIGURE 24.

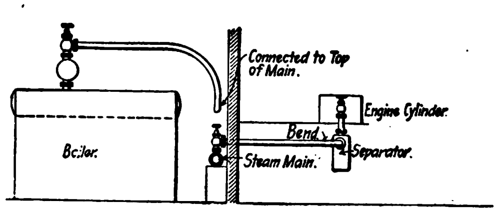



Fig. 25.

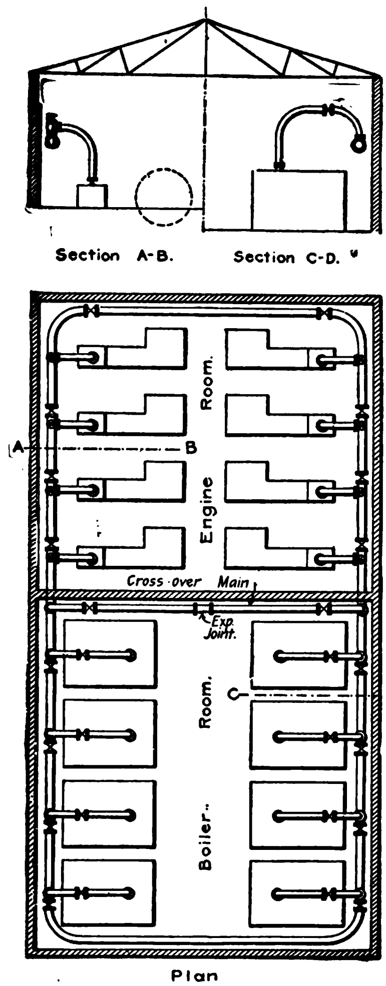



Fig. 26.

When the steam piping in the engine room is run in the basement, the arrangement shown in Fig. 25 can be resorted to. There the main is close to the floor of the boiler room. It can be supported on piers or wall brackets. The stop valve on the boilers is placed on the nozzle and a pipe with a long bend drops into

the top of the main. The branch to each engine is run from an angle stop valve on the top of the main to a separator near each engine cylinder. This system is as unlikely to have accidents occur to it as is the arrangement shown in Fig. 24. An excellent example of this kind is shown in Plate 7.

Sometimes the steam piping is put in in duplicate, the two systems dividing at a double nozzle or Y on the boilers, and converging at a similar Y close to and connecting with the throttle valve of the engine. Duplicate systems were used much more frequently in the early days of electric power-house construction than they are at the present time. In fact, the opinion is fast becoming universal that a duplicate system is an expense that is unnecessary with the arrangement of boilers and engines and piping shown in Fig. 24 or 25.

It is the custom in the latest work to subdivide the power house into complete and independent units. The plans of the Lincoln Wharf power house of the Boston Elevated Railway in Plate 1 and Fig. 3 show how the piping is subdivided. By closing valves in the main each unit is entirely independent.

If the power station has the engine and boiler rooms placed end to end, as in Fig. 26, the arrangement of piping shown there is probably the safest. It is arranged on the ring or loop system, and valves are so placed that if an accident occurs the damaged section may be cut out and the steam carried around through the system in the opposite direction. In the station shown an expansion joint is placed in the cross connection. The cross-over pipe might, perhaps, be omitted. In a station of any size the expansion in the mains running lengthwise of the engine and boiler rooms could be taken care of by anchoring the mains at their middle points, so that the expansion would be divided equally between the two ends of the mains.

**Exhaust Piping for Condensing Plants.** — A frequent method of running exhaust pipes in condensing plants is shown in Fig. 27. Each engine is supposed to be provided with an independent condenser and air pump. Two exhaust-pipe branches are shown, one branch dropping into the condenser and the other branch, which contains a relief valve, leading to the atmosphere. If advantageous to do so, the free-exhaust pipes can be connected to a single pipe leading to the atmosphere. The purpose of the atmospheric connection is to provide means for allowing the steam

to escape in case something happens to the condenser to prevent it from working. The relief valve is nothing more than a large check valve that is closed by the pressure of the atmosphere on one side, when a partial vacuum exists upon the other. Sometimes several engines exhaust into one condenser. The general arrangement can be the same. Exhaust piping for condensing plants should have flanged fittings most carefully put together. A leak through a very small hole will greatly affect the vacuum and the efficiency of the engine. The principal point to be looked after is to have the alignment of the pipe such that there is absolutely no chance for water to lodge in the piping system, for the reason that the water might be sucked back in the engine cylinder and destroy it, as has frequently occurred. It is practically impossible to drain the exhaust pipe of a condensing engine, except toward the condenser, as the system is under a partial vacuum. One of the most important points is to have very generously proportioned exhaust pipes with as direct a run to the condenser and as few bends in the pipe as possible.

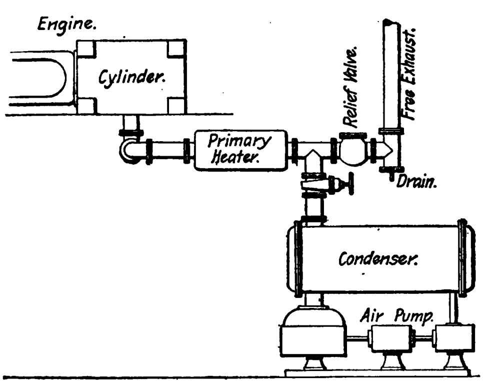



Fig. 27.

**Piping between Cylinders of Compound Engines.** — It is sometimes the custom in installing a cross-compound engine to arrange the piping between the cylinders so that high-pressure steam may be admitted to the low-pressure cylinder as well as the high-pressure, and to provide the necessary exhaust pipes so that both cylinders may be used at the same time as simple engines. If the engine is to be run condensing, it is sometimes so piped that

the high-pressure cylinder may run as a simple noncondensing engine, while the low-pressure cylinder may run as a simple condensing or noncondensing engine, as may be desired. Designers have gone so far as to provide for running high-pressure cylinders as a simple condensing engine and the low-pressure cylinder as a simple noncondensing engine, although ordinarily intended to operate as an engine of the cross-compound condensing type. Various arrangements for accomplishing the purpose mentioned are shown in Plate 3 and in Fig. 18.

**Exhaust Piping for Noncondensing Plants.** — In noncondensing plants the exhaust steam can be carried long distances for heating or for manufacturing purposes, provided the pipes carrying it are large enough. When exhaust steam is thus utilized, the pipe carrying the exhaust steam from the engines to the outer air, which is known as the atmospheric exhaust, or the free-exhaust pipe, is provided with a back-pressure valve, the function of which is to preserve a sufficient pressure in the exhaust piping to cause the exhaust steam to flow through a system of pipes, also connected with this free-exhaust pipe, to the places where it

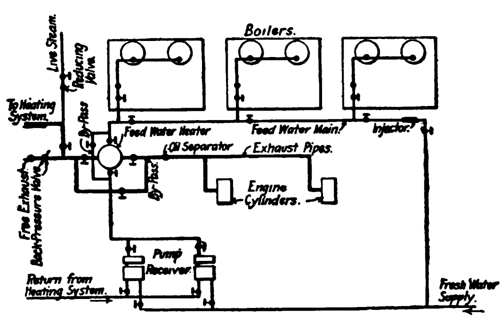



Fig. 28. Exhaust and Feed Piping for Noncondensing Plant.

is to be used. These back-pressure valves are designed so as to open when the pressure in the exhaust system exceeds a certain amount, and thus allow sufficient steam to escape to reduce the pressure to that desired. It sometimes happens, when exhaust steam is used for heating or for manufacturing purposes, that the supply is not sufficient to meet the demand, and if this is likely to occur, it is the custom to run a live-steam pipe from the high-pressure piping to the exhaust piping, and to place in this

connecting pipe a reducing valve which automatically opens and allows live steam to enter the exhaust system when the pressure in the latter falls below that which it is desired to maintain.

When a system for utilizing exhaust steam is employed, it is frequently arranged in the manner shown in diagram by Fig. 28. A grease separator should be provided to remove as much oil as possible from the exhaust steam, after which the steam is passed through the feed-water heater. The arrangement shown is designed for a feed-water heater of the closed type. From the heater the pipe branches, one branch supplying steam for heating or for a manufacturing purpose — this branch furnished with a high-pressure connection run from the boiler and containing a reducing valve. The other branch leads to the atmosphere, and it is provided with the back-pressure valve. The top of the free-exhaust pipe should have an exhaust head for intercepting the moisture that is blown out of the pipe by the exhaust steam. This head should have a drain pipe to a sewer or to waste.

Most power plants have a basement under the floor of the engine room, and in steam plants for buildings the exhaust piping is frequently run in covered trenches. When the exhaust piping is below the floor, a satisfactory method of connecting the feed-water heater is shown in Fig. 29. The steam may pass through the heater or around it through the by-pass by properly adjusting the valves. It is quite common to connect a heater on the induction principle by providing a single connection and relying upon the pressure of the exhaust steam to fill the heater and drive out the air through a small air valve provided for the purpose. The double connection to a heater is much to be preferred.

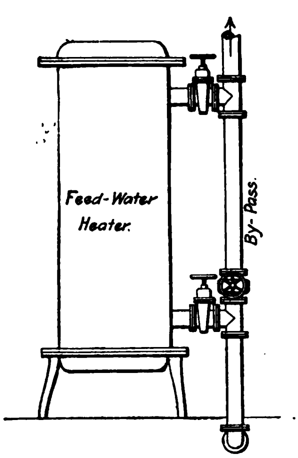



Fig. 29.

**Piping for Building Power Plants.** — Fig. 30 shows in diagrammatic form the principal piping in a building containing about 900 horse-power in boilers for which the writer acted as consulting engineer for the steam work, and is typical of other installations. High-pressure steam is supplied to three engines

driving the electrical generators, to two boiler feed pumps, to three vacuum pumps, to three drip pumps, to two house-tank pumps, to a compressor for a refrigerating plant, to kitchen utensils requiring steam, and to the heating system.

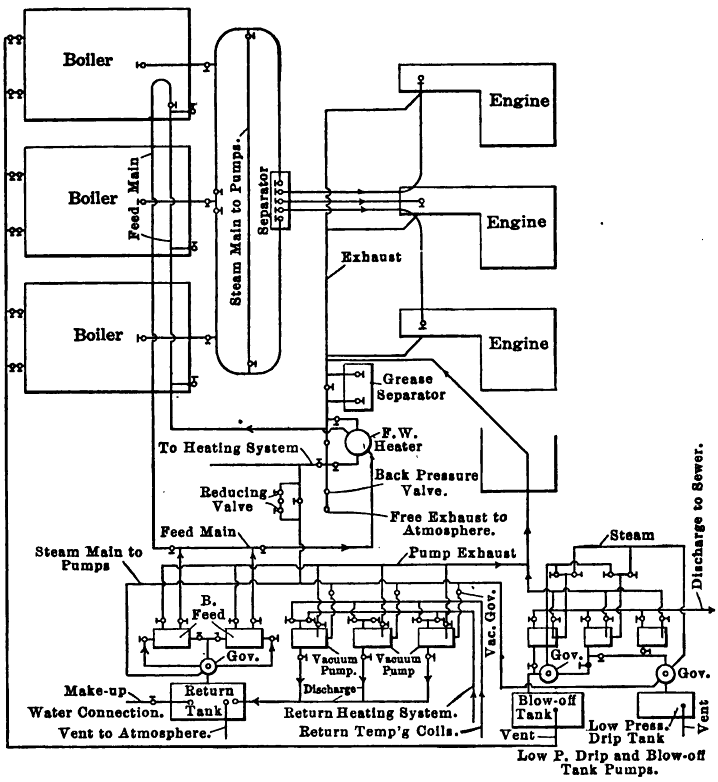



Fig. 30.

There are two valves in the connection to each boiler also in the connection to each engine and pump. The valves on the boilers are of the automatic stop-and-check pattern which will close when an excessive flow out of the boilers due to a break in the steam-distributing system occurs, also when a break occurs in the boiler causing a flow of steam in the reverse direction, i.e., from the main into the boilers. To prevent water from entering the engine cylinders and to prevent vibration due to the intermittent flow of steam to the engines a receiver about 10 feet

long and 3 feet in diameter is placed in the engine room and from this a separate branch is run to each engine. It would have been preferable to have placed a separate receiver on the cylinder of each engine, but the limited head room in the engine room prevented the use of a receiver of sufficient volume.

The exhaust steam from all engines and pumps is collected in a main and then passed through a grease separator from which a free or atmospheric exhaust pipe runs to the roof. Close to the grease separator are branches from the exhaust main to the feed-water heater, to the hot-water generator, and to the heating system and between these connections and the atmosphere a back-pressure valve is placed in the exhaust main which can be adjusted to maintain any desired pressure on the exhaust main up to this valve, so as to force the steam into the heaters and into the heating system. The pressure in a well-designed system should not exceed from one to two pounds pressure above the atmosphere and a lower pressure should suffice where a vacuum heating system is used than would be required without it.

The building is heated by direct radiation and a considerable volume of tempered air is supplied for ventilating purposes. Exhaust steam, supplemented by live steam through an automatically acting reducing valve, is used in the direct radiators and tempering coils; and to insure a thorough and noiseless circulation at low pressures, a vacuum system of heating is used; that is, vacuum pumps are attached to the returns from the heating system to suck the air and condensation from the system and maintain a slight vacuum on the return lines. This vacuum is maintained at a predetermined amount by means of a vacuum governor controlling the steam supply to the pumps. In this case the air tempering coils have a main return independent of return from the direct heating system and they are so piped that either return may be connected to any one of the three vacuum pumps, so that by having three pumps there will always be a spare pump should one break down.

The boiler-feed pumps, of which there are two, draw water either from a main return tank, from the city supply direct, or from the water storage tank for the building, and force the water into the boilers either through the feed-water heater or direct. Either pump is sufficient in size so there is a spare pump. Both are governed by a pump governor of the float type attached to

the return tank. In this plant, and in most others, the amount of steam exhausted by the engines and pumps is in excess of the amount of steam required to heat the building and this excess passes to the atmosphere by way of the free or atmospheric exhaust. When this occurs it is manifest that the water returning from the heating system and the drips to the return tank will not be sufficient to supply the boilers, hence fresh water, or " make-up water " as it is called, is necessary. This is arranged for by running a cold-water pipe with valve in a convenient location in the boiler room to the return tank. Upon the opening of this valve by the boiler attendant the water level in the tank and governor, which are cross connected or equalized, rises so as to raise the float in the governor and so start the feed pump. The movement of the float up or down closes or opens a valve in the steam supply to the pump.

The vacuum pumps and clean drips discharge into the return tank. The vacuum pumps for the heating system discharge more or less air and to relieve the return tank of all pressure a 4-inch vapor line running to the roof is provided. The drips running to the return tank consist of the condensation that occurs in the high-pressure mains, in the hot-water generator, in the steam separator, and in the feed-water heater. The grease separator is placed between the engines and the connections to the feed-water heater and the hot-water generator, and, as the grease separator may be out of use occasionally, means is provided for turning all the high-pressure drips into the low-pressure or the dirty drip system. All drips from the high-pressure mains and branches are provided with traps; also the feed-water heater and hot-water generator drips. The heater and hot-water generators are set at such a level as to drain by gravity into the return tank.

The low-pressure or dirty drips consist of drips in the exhaust mains and branches, grease separator, cylinder drains, etc., and these are run to a low-pressure drip tank which is about 3 by 4 feet in size located in a pit so as to be well below the level of the lowest point to be drained. The tank is relieved of pressure by a 3-inch vapor line to the roof, and a pump controlled by a governor of the float type maintains a constant level in the drip tank and discharges into the main house sewer outside of the plumber's trap, this being done to prevent the vapor from the

pump discharge from backing up into the house drainage system. Located in the same pit with the low-pressure drip tank is a blow-off tank into which the blow-off mains from the boilers connect. This is also vented to the roof by an independent pipe and is drained by a pump and governor also discharging into the sewer. In some cities an ordinance prohibits the discharge of steam or vapor-bearing water into the sewers and to overcome this it is customary to place within the blow-off tank and low-pressure drip tank a cooling coil of brass pipe through which the cold water supplied to the feed pumps is passed.

While the plant shown in Fig. 30 and described above is typical of others there are many modifications of detail in practice. If an open type of feed-water heater is used the vacuum pumps would ordinarily be arranged to discharge into a separating tank perhaps 18 inches in diameter and 4 or 5 feet in length, and this would be vented to the atmosphere. The return tank would be omitted and the returns led directly into the feed-water heater near the bottom. The fresh water would be automatically supplied to the top of the heater by a float within. The separating tank would be elevated considerably above the heater, and it would be connected to the heater so that the water discharged by the vacuum pumps would pass from the separating tank to the heater by gravity. The pipe connecting them would contain what is known as a loop seal to prevent the exhaust steam in the heater from entering the separating tank and escaping to the atmosphere by way of the vapor pipe from this tank. The height of the column of water in the loop seal determines the maximum pressure therefore that can be carried in the heating system.

Still one other method of piping exhaust-steam heating plants is to connect the returns from the direct-heating system into a return tank vented into the heating system drained by boiler-feed pumps perhaps with a governor with all high-pressure drips run to an independent tank pump and governor, the drip pump discharging into the feed main between the boiler-feed pumps and the governor. If a vacuum system of heating had not been used in the plant illustrated in Fig. 30, the returns from the direct-heating system could have been independently trapped into the return tank and the high-pressure drips as well, if the return tank was provided with a free vent to the atmosphere.

**Care of Drips.** — The drainage from any part of the piping system is valuable on account of the value of the water, and the heat in the water that would be lost if the condensation was allowed to go to waste. By condensation is meant water due to condensation in the pipes. Sometimes the saving that is due to returning high-pressure condensation to the boilers is not sufficient to warrant the expenditure for the apparatus necessary to do this, but that is a point which must be determined for each case. Condensation from exhaust-steam pipes can sometimes well be allowed to go to waste, for the reason that it generally contains more or less oil; and the chance of this doing injury to the boilers is apt to be too great for the saving that would follow its utilization. If it is used, the water should be filtered or an efficient grease separator should be used.

There are various methods of returning high-pressure condensation to the boilers, and the most common are the Holly system and the "steam loop," both patented systems controlled by Westinghouse, Church, Kerr & Company, by means of the automatically-governed steam pump and receiver and by means of a return trap. With the pump and receiver all high-pressure drip pipes, from separators, steam jackets on the cylinders, from reheating receivers placed between the cylinders of compound and triple-expansion engines, and all other points from which water is drained can be connected to a main leading to an automatic pump and receiver. Each drain pipe should have a steam trap, if there is any chance of their being under different pressures. The automatically-governed pump can discharge the water into the boiler-feed pipe between the boiler-feed pump and the boilers. With very long steam pipes which have to be drained at several points, the drainage can be drawn off by a steam trap discharging into a return pipe leading to an automatic pump and receiver at the boiler house. If the inclination of the steam pipe is such that the water will not run back by gravity in the return pipe to the boiler house, the drip can be carried to the lowest point in the system and an automatic pump and receiver, operated by steam from the main pipe, can be located there and used to pump the water up hill and back to the boiler house; that is, of course, provided the saving will warrant this arrangement. All high-pressure drip lines should be thoroughly covered with a nonconducting covering.

Low-pressure drips contain the condensation drawn from pipes carrying exhaust steam, steam condensed in feed-water heaters of the closed type, the drip from engine and pump cylinders, and the drip from grease separators. All of this condensation should be thrown away on account of the grease that it contains, unless some form of oil filter or an efficient grease separator is used. All low-pressure drips should be trapped independently into a drip main which can be led to any convenient place, such as the sewer, the pipe carrying off the discharge from the condenser, etc.

**Feed-water Piping.** — Feed-water piping is sometimes put together with screwed and sometimes with flanged fittings, and these are either of brass or cast iron. The pipe should be of brass, iron-pipe size. It is better to use elbows with a long radius to reduce friction or, what is better still, pipe bends of long radius wherever possible. Gate valves should be used instead of globe valves, for the same reason, excepting feed-controlling valves on boilers which should be of the globe type.

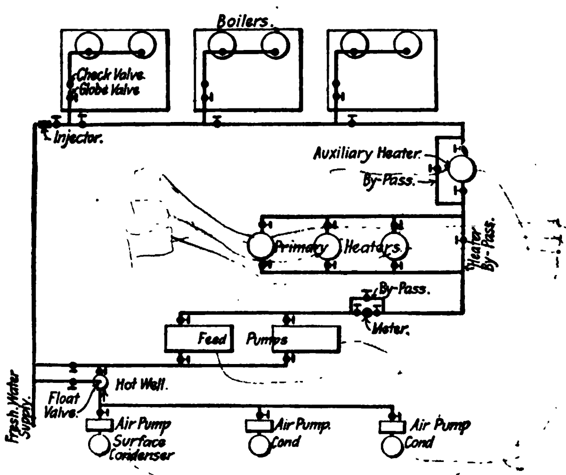



Fig. 31. Feed Piping and Condensing Plant.

An arrangement for feed-water piping for a typical plant equipped with surface condensers is shown in diagram by Fig. 31. The plant is supposed to contain three engines, each with an independent air pump and surface condenser, two boiler-feed pumps, a primary heater in the exhaust pipe of each engine, between the engine and surface condenser, and a single auxiliary

heater of the closed type receiving the exhaust steam of the air pumps and the boiler-feed pumps. The steam condensed in each condenser is drawn therefrom by the air pumps and forced into a hot well, from which the boiler-feed pumps draw their supply. For occasionally replenishing the water from the condensers and supplying fresh water when needed, a fresh-water pipe is led to the hot well, and its supply can be controlled by a float valve or ball cock, so that fresh water will flow in the hot well in case the boiler-feed pumps draw water faster than it is discharged from the air pumps, an occurrence that is unlikely to happen. The boiler-feed pumps are in duplicate. It is a good investment to put a water meter in the feed line, and one capable of measuring hot water should be used. The meter should have a by-pass and be located at the pressure side of the pump, as shown. From the feed pumps the water is forced through the primary heaters and then through the auxiliary heater to the boilers. If the hot well is not provided with sponges or some material arranged to filter the oil from the air-pump discharge, a cloth filter should be placed between the feed pumps and the boilers. Each heater should have a by-pass, and in the pipe leading to each heater two valves should be placed, one used as a stop valve and the other as a regulating valve, which should be adjusted when the plant is started, so that approximately equal amounts of water will pass through the heaters. It involves a little more complication to do this, but it insures each heater doing its full work.

The arrangement shown in Fig. 31 provides for the use of a heater of the closed type between the engine and condenser, for of course an open heater could not be used in such a situation. It is not often the custom to place a heater between the engine and condenser, and where that is not done an open heater might be used to receive the exhaust steam of the auxiliaries, and if so, the arrangement of the feed-water piping would have to be different from that shown. If the plant shown diagrammatically in Fig. 31 was provided with jet condensers the boiler-feed pumps might be arranged to draw their supply from the air-pump discharge, as well as from the source of the fresh-water supply and pump it through the heaters to the boilers. If ample feed water at low cost was available the air-pump discharge would probably be allowed to go to waste because of the grease in it and the pumps connected to draw fresh water. If the latter was under

pressure the water could be passed to an open heater first and the pump arranged to draw the water from it. This arrangement combined with an economizer is shown diagrammatically in Fig. 32. If a surface condenser was used the air-pump discharge

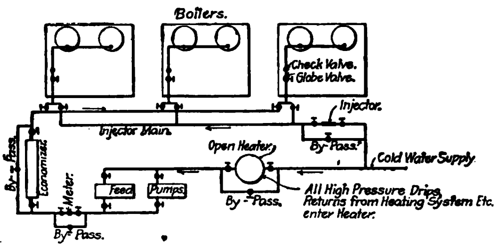



Fig. 32. Feed Piping with Open Heaters.

might be led to an open hot well, arranged to filter the water by means of sponges or excelsior, etc., located at a higher elevation than the heater so the water would flow to the heater by gravity and from the heater to the feed pumps at a lower elevation than the heater so that the pump suction would be under pressure. If the filter was of the closed type containing cloth or similar material it would have to be placed between the pumps and the boilers. High-pressure drips could be trapped into the heater, returns from the heating system could be connected with it, and fresh cold water could be added to the heater by a float valve when the demand of the pumps was in excess of the water entering the heater from the condensers, heating system, drips, etc. The boiler-feed pumps would have to pump hot water with this arrangement, hence they should be constructed to do this. The pumps could deliver water to the boilers direct or first through an economizer where the water would be heated further by the waste gases from the boiler. An economizer is placed between the pumps and the boilers. Arrangements should be made to by-pass all heaters, economizers, meters, etc.

If a live-steam purifier is used, which is a heater in which the feed water is exposed to the action of steam at full boiler pressure to precipitate scale forming salts in the purifier instead of having them precipitated in the boiler, the purifier is elevated above the boilers and receives the feed water after it has passed through all other heaters; and it is connected so that water will flow from it to the boilers by gravity.

In Fig. 31 one end of the feed-water main in the boiler room is provided with an injector connected to the fresh-water supply. It would be a better arrangement, perhaps, to have an independent main from the injector, the main connecting with the feed pipe to each boiler through stop valves so that water from the injector or the feed pump could be supplied to any boiler independent of the others, as shown in Fig. 33.

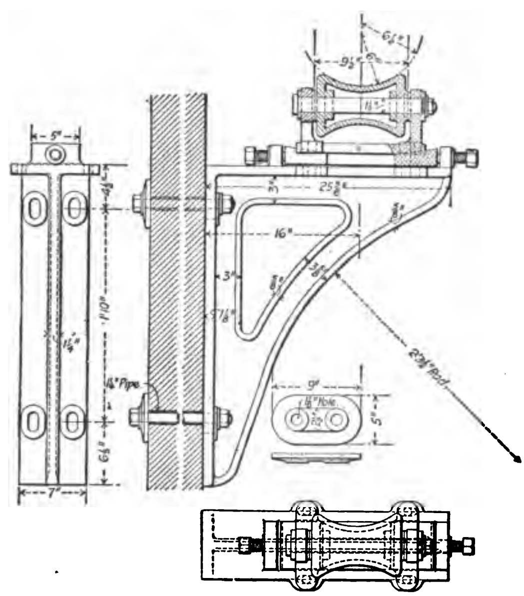


Fig. 33. Pipe Bracket designed by Sheaff & Jaasted.

The method of running feed mains in a boiler room varies. Horizontal tubular boilers are frequently set with a number of boilers in one battery. Sometimes, therefore, the feed main is run along the fronts of the boilers just above the fire doors. Water-tube boilers are generally set two in a battery, so that if a pipe extends across the front it blocks the passageway between the batteries. With this type of boiler the feed main is sometimes run in a basement below the boiler room. Again, the mains are run on top of the setting. In the latter event a branch from the main should extend to the boiler front, at an elevation low enough so that the regulating valve, which should be of the globe type, may easily be reached by the boiler attendant. A pref-

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erable arrangement is to keep the feed piping well overhead and provide an extension to the feed-valve spindle extending down to a convenient height above the floor. A check valve should be placed in the pipe, as shown.

Feed-water piping for noncondensing plants with the closed type of feed-water heater is shown in diagram in Fig. 28. It is supposed that exhaust steam is used for heating and that the condensation is returned to the boilers. This may be brought back to one of the boiler-feed pumps, which can be connected to a return tank receiving the condensation in the heating system. The pump draws water from the tank and forces it through the feed-water heater to the boiler. A fresh-water pipe is led to the return tank and the supply is controlled by a float valve. If a feed-water heater of the open type is used, the condensation from a heating system is carried back to the heater, and usually enters it at the side or bottom and thus does away with the automatic pump and receiver. The boiler-feed pump draws its supply from the heater and forces the water to the boilers. Fresh water under pressure, controlled by a float valve or ball cock in the heater, is supplied to the heater at the top, and it falls to the bottom in direct contact with the exhaust steam and is thus heated. Cold water is only supplied when the boiler-feed pump draws the water from the heater so fast that the water surface falls below a certain level.

In electric stations it is frequently the practice, for safety's sake, to use a duplicate feed main from the pump to the boilers. With such an arrangement it is possible to test a boiler or group of boilers. If the steam piping is arranged so that one or more boilers can supply any one engine independently of the others, the duplicate boiler-feed main is of considerable value, for with such a duplicate system the steam used by an engine can be measured at any time and the condition of the engine determined. Boiler-feed pumps should always be in duplicate.

**Blow-off Piping.** — Blow-off piping is subjected to sudden changes of temperature and stress and special care should be taken to guard against excessive expansion strains. It should be put together with extra heavy cast-iron fittings except on the blow-off from horizontal return tubular boilers, and in this case when the blow-off is connected to the bottom of the boiler at the rear, with an elbow in the pipe exposed to the fire, this fitting

should be of malleable iron. Fittings and valves outside of the boiler setting should be flanged. Each boiler blow-off should have two valves or a blow-off valve with a plug cock inside of the valve. Many water-tube boilers are provided by the makers with an angle type of blow-off valve, and when this is used the cock can be placed between it and the boiler. Single blow-off valves are very apt to leak, due apparently to the cutting of the valve seat and disk by the scale blown from the boiler. With two valves this trouble is largely overcome.

In buildings in cities where an ordinance prevents the blowing of boilers into the sewer the blow-off is frequently led to a tank vented to the atmosphere, the tank being provided with a coil of pipe through which cold water used for the boiler feed or for the hot-water supply of the building is circulated, for the purpose of cooling the water blown from the boilers before it is run into the sewer. Sometimes a cold-water connection is run to this tank and cold water is used for cooling. As the water blown from the boilers is frequently at a temperature way above the temperature at which water under atmospheric pressure evaporates, a certain part of the water blown from boiler will evaporate into steam, and means should be provided for disposing of this steam without injury.

For large power plants the blow-off can be run into a specially constructed reservoir or cesspool roofed over and vented to the atmosphere. The blow-off main from the boilers can be connected at the top, and the discharge for the water can be through an overflow pipe extending from a point near the bottom of the cesspool to a point near the top then running horizontally to a sewer or to waste. With this arrangement the cooler water will be at the bottom and be drawn off first.

# CHAPTER VIII.

## MATERIALS FOR PIPING, PIPE SIZES, SEPARATORS, AND OILING SYSTEMS.

**Kind of Pipe.** — " Standard " sizes of pipe of steel or wrought iron are usually used for steam piping. Table 20 contains the various dimensions of pipe, those up to and including 10-inch

TABLE 20. — DIMENSIONS OF STANDARD-WEIGHT PIPE.

| Inside diameter. | Actual outside diameter. | Thickness. | Actual inside diameter. | Length of pipe per square foot of outside surface. | Inside area. | Length of pipe containing one cubic foot. | Weight per foot. | Discharge per minute, velocity 6000 feet per minute, steam 100 pounds gauge pressure. | Discharge per minute, velocity 6000 feet per minute, steam 150 pounds gauge pressure. |
|---|---|---|---|---|---|---|---|---|---|
| Ins. | Ins. | Ins. | Ins. | Feet. | Ins. | Feet. | Lbs. | Lbs. | Lbs. |
| 1 | 1.315 | 0.134 | 1.048 | 2.903 | 0.8627 | 166.9 | 1.670 | ........ | ...... |
| 1¼ | 1.66 | 0.140 | 1.380 | 2.301 | 1.496 | 96.25 | 2.258 | ........ | ...... |
| 1½ | 1.90 | 0.145 | 1.611 | 2.01 | 2.038 | 70.65 | 2.694 | ........ | ...... |
| 2 | 2.375 | 0.154 | 2.067 | 1.611 | 3.355 | 42.36 | 3.600 | ........ | ...... |
| 2½ | 2.875 | 0.204 | 2.468 | 1.328 | 4.783 | 30.11 | 5.773 | ........ | ...... |
| 3 | 3.50 | 0.217 | 3.067 | 1.091 | 7.388 | 19.49 | 7.547 | 80 | 113 |
| 3½ | 4.00 | 0.226 | 3.548 | 0.955 | 9.887 | 14.56 | 9.055 | 107 | 151 |
| 4 | 4.50 | 0.237 | 4.026 | 0.849 | 12.730 | 11.31 | 10.66 | 138 | 194 |
| 4½ | 5.00 | 0.247 | 4.508 | 0.765 | 15.939 | 9.03 | 12.34 | 173 | 243 |
| 5 | 5.563 | 0.259 | 5.045 | 0.629 | 19.990 | 7.20 | 14.50 | 218 | 305 |
| 6 | 6.625 | 0.280 | 6.065 | 0.577 | 28.889 | 4.98 | 18.767 | 315 | 442 |
| 7 | 7.625 | 0.301 | 7.023 | 0.595 | 38.737 | 3.72 | 23.27 | 422 | 592 |
| 8 | 8.625 | 0.322 | 7.982 | 0.444 | 50.039 | 2.88 | 28.177 | 545 | 764 |
| 9 | 9.625 | 0.344 | 9.001 | 0.394 | 63.633 | 2.26 | 33.70 | 694 | 974 |
| 10 | 10.75 | 0.366 | 10.019 | 0.355 | 78.838 | 1.80 | 40.06 | 872 | 1223 |
| 11 | 12.00 | 0.375 | 11.25 | 0.318 | 98.942 | 1.455 | 45.95 | 1079 | 1514 |
| 12 | 12.75 | 0.375 | 12.000 | 0.293 | 116.535 | 1.235 | 48.98 | 1275 | 1788 |
| 13 | 14.00 | 0.375 | 13.25 | 0.273 | 134.582 | 1.069 | 53.92 | 1468 | 2060 |
| 14 | 15.00 | 0.375 | 14.25 | 0.254 | 155.968 | .923 | 57.89 | 1702 | 2385 |
| .... | 16.00 | 0.375 | 15.25 | 0.238 | 177.867 | .809 | 61.77 | 1940 | 2722 |
| .... | 18.00 | 0.375 | 17.25 | 0.212 | 225.907 | .638 | 69.66 | 2462 | 3452 |
| .... | 20.00 | 0.375 | 19.25 | 0.191 | 279.720 | .515 | 77.57 | 3050 | 4275 |
| .... | 22.00 | 0.375 | 21.25 | 0.174 | 354.66 | .406 | 85.47 | 3870 | 5425 |

pipe being the Briggs standard. The other sizes are in common use. As ordinary merchant pipe may vary in thickness from

the standard, "full-weight" pipe should be asked for. Full-weight pipe may vary 5 per cent from the standard thickness. In the connection between a boiler and a steam main, or the main and the engines, long bends made of pipe are an advantage, for several reasons. First, they reduce the friction very much; second, their use reduces the number of joints likely to leak; third, such a connection is very much more flexible than one composed of two straight pieces of pipe connected by an elbow. Their greater flexibility is of great advantage in taking care of expansion after the piping is in place, and, furthermore, they are much easier to connect when erecting the piping. No matter how much care is taken in facing the flanges off square, it almost always happens that the flanges of the boiler nozzles are not in perfect alignment, or exactly horizontal, so that a considerable strain is introduced in the piping in forcing the abutting flanges to a seat. It is much better to make the bends in the piping of steel or wrought iron than copper, although the latter has been used to some extent. At the temperature the copper is subjected to in brazing the joint, the fibrous nature that copper acquires in rolling is destroyed and a serious reduction of its tensile strength and ductility results. Mr. James B. Berryman of the Crane Company states that unless the bends are of very short radius they are generally made of standard pipe for pressures of 125 pounds or less, full-weight pipe up to 175 pounds, and extra-heavy pipe for higher pressures.

**Size of Steam Pipes.** — It has been the custom to so proportion steam-supply pipes for engines that the maximum velocity of steam flowing through them will be about 6000 feet per minute. The pipe size can be obtained by assuming that the area of the steam pipe in square inches multiplied into the maximum velocity of steam in feet per minute is equal to the piston area in square inches multiplied into the piston speed in feet per minute. If the cut-off is at one-third stroke the average velocity of steam would then be about 2000 feet per minute.

Friction would cause a considerable drop in pressure, with small sizes of pipe, when proportioned by this rule. Some experiments upon the flow of steam in pipes in a paper in Volume XX, Transactions of The American Society of Mechanical Engineers, by Professor R. C. Carpenter, give some data that is of value. The tests were made upon 1-, $1\frac{1}{2}$-, 2- and 3-inch pipes with

lengths varying from 90 to 250 feet. The formula derived from the experiments checks very closely with the results obtained

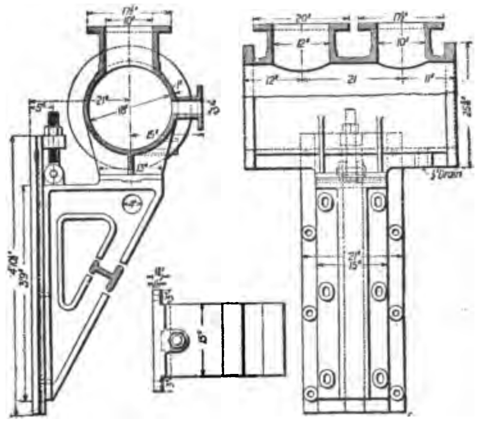


Fig. 34. Pipe Bracket.

by M. Ledoux, who experimented with pipes varying from 1.85 inches to 3.94 inches in diameter and with lengths varying from 328 feet to 1082 feet.

Professor Carpenter uses the formula

$$P = \frac{1}{20.663} K\left(1 + \frac{3.6}{d}\right)\frac{W^2L}{Dd^5},$$

in which $P$ equals the loss of pressure in pounds per square inch, $d$ the diameter of the pipe in inches, $W$ the flow of steam in pounds per minute, $w$ the flow in pounds per second, $D$ the density or weight per cubic foot, $L$ the length of pipe in feet and $K$ a constant taken, as a result of the experiment, at .0027. Table 21, which was calculated by Mr. E. C. Sickles, is based upon the formula and was deduced by making every factor constant except the diameter, length of pipe, and discharge. The left-hand vertical column of the table contains the diameters ($d$) of the pipes, and the top horizontal column the length ($L$) in feet, while the body of the table gives values of ($W$) the pounds discharged per minute. Professor Carpenter explains the table as follows:

"Thus, for instance, a 10-inch pipe 250 feet long will deliver 712 pounds of steam per minute with a drop of one pound in pressure, if there exists an average absolute pressure of 100

pounds; or, if all other conditions hold except the length of pipe, which varies, it may be seen that for 100 feet the discharge

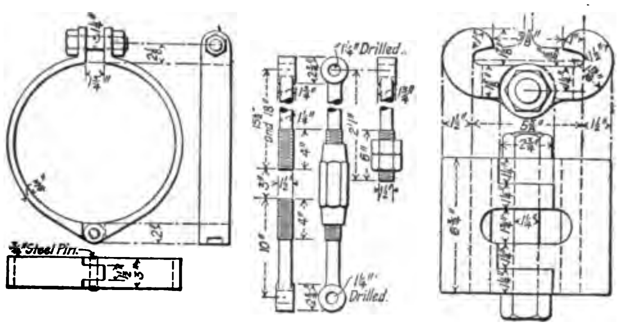


Fig. 35. Pipe Support designed by Dean & Main.

is 1126 pounds, for 400 feet 563 pounds, and so on for any number of feet given in the table.

" If any intermediate length of pipe is used other than those given in the table, the discharges given by the table may be corrected by consideration of the fact that the weight of discharge is inversely proportional to the square root of the length of the pipe.

" To meet the conditions where other average absolute pressures than 100 pounds exist, and higher drops than one pound are assumed, it is only necessary to use suitable factors which are calculated by means of the fundamental formula, and graphically represented by Curves 1 and 2, Fig. 36.

" As an illustration of the use of the tables and curves, suppose it is desired to find what size pipe will be required to deliver 1000 pounds of steam per minute a distance of 1000 feet, the initial pressure of the steam being 157.5 pounds, and the final 152.5 pounds by gauge. Solution — It will be best to reduce all conditions to those of the tables and find the discharge, and from this the size of the required pipe. Looking at Curve 2, we find the factor of discharge for a 5-pound drop is about 2.23 times that for a 1-pound drop. Therefore, dividing the required discharge of 1000 pounds by 2.23, we have about 450 pounds discharge for a 1-pound drop.

" Again, the average pressure is 155 + 15, or 170 pounds absolute, and from Curve 1 it may be found the factor of discharge is

TABLE 21. — FLOW OF STEAM IN PIPES. DELIVERY IN POUNDS PER MINUTE BY PIPES OF GIVEN DIAMETERS AND LENGTHS. (ABSOLUTE PRESSURE 100 POUNDS — DROP IN PRESSURE ONE POUND.)

| Diameter in inches. | Length in feet. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 50 | 100 | 150 | 200 | 250 | 300 | 350 | 400 | 500 | 600 |
| $\frac{3}{4}$ | 1.55 | 1.10 | .898 | .700 | .698 | .633 | .589 | .550 | .491 | .450 |
| 1 | 3.10 | 2.20 | 1.79 | 1.55 | 1.39 | 1.26 | 1.17 | 1.10 | .982 | .900 |
| $1\frac{1}{4}$ | 6.90 | 4.88 | 3.93 | 3.44 | 3.08 | 2.81 | 2.62 | 2.44 | 2.18 | 1.96 |
| $1\frac{1}{2}$ | 9.29 | 6.52 | 5.83 | 4.62 | 4.14 | 3.77 | 3.50 | 3.26 | 2.91 | 2.67 |
| 2 | 21.6 | 15.2 | 12.50 | 10.8 | 9.82 | 8.82 | 8.37 | 7.63 | 6.81 | 6.23 |
| $2\frac{1}{2}$ | 35.9 | 25.4 | 20.8 | 17.9 | 16.2 | 14.62 | 13.6 | 12.7 | 11.3 | 10.4 |

| Diameter in inches. | Length in feet. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 100 | 250 | 400 | 550 | 700 | 850 | 1000 | 1300 | 1600 | 1750 |
| 3 | 46.0 | 29.2 | 23.0 | 19.6 | 17.4 | 15.8 | 14.5 | 12.7 | 11.5 | 11.0 |
| $3\frac{1}{2}$ | 69.5 | 44.6 | 34.7 | 29.6 | 26.3 | 23.8 | 22.0 | 19.2 | 17.4 | 16.6 |
| 4 | 97.6 | 61.8 | 48.8 | 41.5 | 36.7 | 33.5 | 30.8 | 27.1 | 24.4 | 23.3 |
| $4\frac{1}{2}$ | 132.9 | 84.1 | 66.45 | 56.6 | 50.1 | 45.6 | 42.0 | 36.75 | 33.2 | 31.8 |
| 5 | 180.7 | 114.3 | 90.3 | 76.9 | 68.2 | 62.1 | 57.1 | 50.0 | 45.2 | 43.2 |
| 6 | 296.5 | 187.4 | 148.2 | 125.8 | 111.8 | 101.8 | 93.7 | 82.2 | 74.1 | 71.0 |
| 7 | 437 | 276 | 218 | 186 | 165 | 150 | 138 | 121 | 109 | 104 |
| 8 | 625 | 394 | 312 | 266 | 236 | 214 | 197 | 173 | 156 | 149 |
| 9 | 853 | 539 | 426 | 363 | 322 | 292 | 269 | 236 | 213 | 204 |
| 10 | 1126 | 712 | 563 | 480 | 425 | 386 | 356 | 312 | 281 | 269 |
| 11 | 1447 | 915 | 723 | 617 | 546 | 496 | 457 | 401 | 361 | 346 |
| 12 | 1887 | 1192 | 943 | 803 | 714 | 648 | 598 | 523 | 472 | 451 |
| 13 | 2238 | 1415 | 1119 | 954 | 846 | 767 | 707 | 620 | 559 | 535 |

| Diameter in inches. | Length in feet. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 100 | 200 | 300 | 500 | 800 | 1000 | 1400 | 1800 | 2000 | 2200 |
| 14 | 2,714 | 1920 | 1567 | 1213 | 959 | 858 | 725 | 639 | 606 | 578 |
| 15 | 3,250 | 2300 | 1873 | 1453 | 1149 | 1028 | 868 | 766 | 726 | 693 |
| 16 | 4,000 | 2830 | 2315 | 1785 | 1413 | 1268 | 1072 | 945 | 895 | 854 |
| 17 | 4,500 | 3210 | 2635 | 2021 | 1591 | 1424 | 1200 | 1061 | 1006 | 962 |
| 18 | 5,211 | 3685 | 3008 | 2330 | 1843 | 1648 | 1393 | 1228 | 1165 | 1111 |
| 19 | 5,992 | 4237 | 3459 | 2680 | 2119 | 1895 | 1602 | 1412 | 1340 | 1278 |
| 20 | 6,839 | 4835 | 3948 | 3059 | 2418 | 2163 | 1828 | 1612 | 1529 | 1458 |
| 22 | 8,743 | 6183 | 5048 | 3910 | 3093 | 2765 | 2337 | 2061 | 1955 | 1864 |
| 24 | 11,308 | 7990 | 6535 | 5065 | 3995 | 3580 | 3023 | 2665 | 2535 | 2415 |

1.248 greater than for 100 pounds absolute. Therefore, dividing 450 pounds by 1.284 we have 350 pounds on the basis of the conditions given in Table 21; and looking under 1000-feet lengths

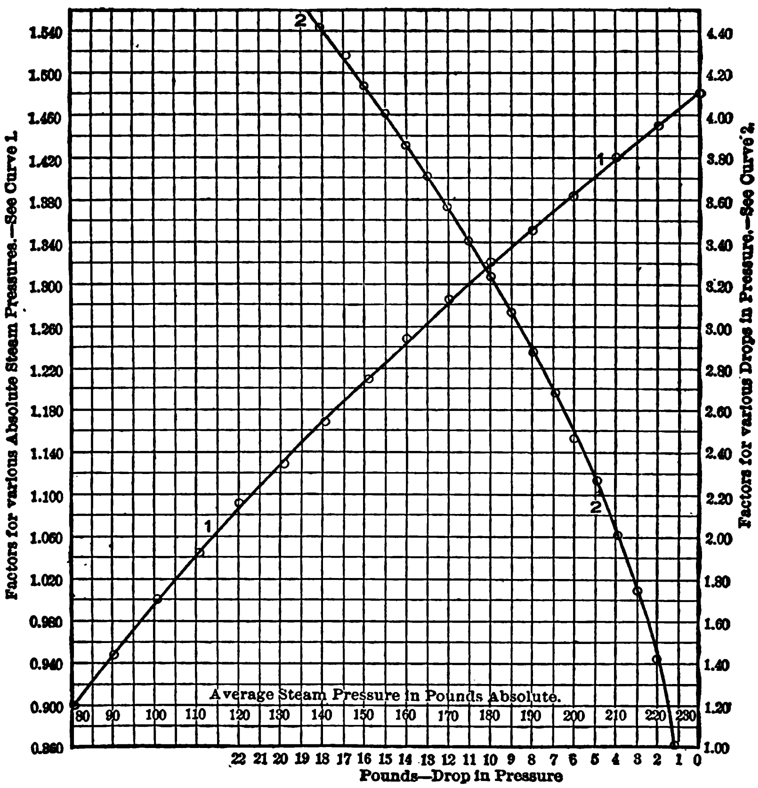



Fig. 36.

for the discharge nearest to 350 pounds, we find a 10-inch pipe discharges 356 pounds per minute; therefore, it would be satisfactory."

**Kind of Fittings.** — There are two kinds of fittings used in steam piping, the screwed and the flanged fittings. The former, as the name implies, are put together with a screw thread, while the flanged fittings are bolted together. Screwed fittings are used to some extent in plants where the steam pressure is low, 80 pounds or under, and sometimes with higher pressures. Adjacent pieces of pipe may be coupled together by means of screw couplings or by means of companion flanges. Screw fittings have the advantage that once made up properly they are not likely to leak, while flanged fittings have to be put together with some kind of gasket which may blow out or require renewal.

Flanged connections between adjacent pieces of pipe and flanged fittings and valves possess the great advantage of being easily taken apart and repaired, and this is so great an advantage that flange construction is much preferred in high-grade work. In screw work it is customary to specify that occasional flanges are to be provided in piping to permit of its being taken down easily. The most used type of flange is shown in Fig. 37. The pipes are screwed into the flanges as shown. On July 18, 1894, committees of the American Society of Mechanical Engineers, of the National Association of Steam and Hot-water Fitters, and of manufacturers of fittings, met and adopted a schedule for the dimensions of flanges and this schedule is known as the A. S. M. E. or the Master Steam Fitters' flange schedule. This schedule is printed in Table 22 below. Flanges dimensioned in accordance with this schedule are considered strong enough for a steam pressure of 100 pounds. For higher pressures, a schedule shown

TABLE 22. — SCHEDULE OF STANDARD FLANGES.

Adopted July 18, 1894, by a committee of the Master Steam and Hot-water Fitters' Association, American Society of Mechanical Engineers and Valve Fitting Manufacturers. Suitable for pressures under 100 pounds per square inch.

| Size of flange, pipe size×diameter. | Diameter of bolt circle. | Number of bolts. | Size of bolts, pressure under 80 pounds. | Size of bolts, pressure 80 pounds and over. | Flange thickness at hub for iron pipe. | Flange thickness at edge. | Width of flange face. |
|---|---|---|---|---|---|---|---|
| $2 \times 6$ | $4\frac{3}{4}$ | 4 | $\frac{1}{2}\times 2$ | $\frac{5}{8}\times 2$ | 1 | $\frac{5}{8}$ | 2 |
| $2\frac{1}{2}\times 7$ | $5\frac{1}{2}$ | 4 | $\frac{1}{2}\times 2\frac{1}{4}$ | $\frac{5}{8}\times 2\frac{1}{4}$ | $1\frac{1}{8}$ | $\frac{11}{16}$ | $2\frac{1}{4}$ |
| $3 \times 7\frac{1}{2}$ | 6 | 4 | $\frac{1}{2}\times 2\frac{1}{2}$ | $\frac{5}{8}\times 2\frac{1}{2}$ | $1\frac{1}{4}$ | $\frac{3}{4}$ | $2\frac{1}{4}$ |
| $3\frac{1}{2}\times 8\frac{1}{2}$ | 7 | 4 | $\frac{1}{2}\times 2\frac{1}{2}$ | $\frac{5}{8}\times 2\frac{1}{2}$ | $1\frac{1}{4}$ | $\frac{13}{16}$ | $2\frac{1}{2}$ |
| $4 \times 9$ | $7\frac{1}{2}$ | 4 | $\frac{5}{8}\times 2\frac{3}{4}$ | $\frac{3}{4}\times 2\frac{3}{4}$ | $1\frac{3}{8}$ | $\frac{15}{16}$ | $2\frac{1}{2}$ |
| $4\frac{1}{2}\times 9\frac{1}{4}$ | $7\frac{3}{4}$ | 8 | $\frac{5}{8}\times 3$ | $\frac{3}{4}\times 3$ | $1\frac{3}{8}$ | $\frac{15}{16}$ | $2\frac{3}{8}$ |
| $5 \times 10$ | $8\frac{1}{2}$ | 8 | $\frac{5}{8}\times 3$ | $\frac{3}{4}\times 3$ | $1\frac{1}{2}$ | $\frac{15}{16}$ | $2\frac{1}{2}$ |
| $6 \times 11$ | $9\frac{1}{2}$ | 8 | $\frac{5}{8}\times 3$ | $\frac{3}{4}\times 3$ | $1\frac{1}{2}$ | 1 | $2\frac{1}{2}$ |
| $7 \times 12\frac{1}{2}$ | $10\frac{3}{4}$ | 8 | $\frac{5}{8}\times 3\frac{1}{4}$ | $\frac{3}{4}\times 3\frac{1}{4}$ | $1\frac{1}{2}$ | $1\frac{1}{16}$ | $2\frac{3}{4}$ |
| $8 \times 13\frac{1}{2}$ | $11\frac{3}{4}$ | 8 | $\frac{5}{8}\times 3\frac{1}{2}$ | $\frac{3}{4}\times 3\frac{1}{2}$ | $1\frac{3}{4}$ | $1\frac{1}{8}$ | $2\frac{3}{4}$ |
| $9 \times 15$ | $13\frac{1}{4}$ | 12 | $\frac{5}{8}\times 3\frac{1}{2}$ | $\frac{3}{4}\times 3\frac{1}{2}$ | $1\frac{3}{4}$ | $1\frac{1}{8}$ | 3 |
| $10 \times 16$ | $14\frac{1}{4}$ | 12 | $\frac{3}{4}\times 3\frac{5}{8}$ | $\frac{7}{8}\times 3\frac{5}{8}$ | 2 | $1\frac{3}{16}$ | 3 |
| $12 \times 19$ | 17 | 12 | $\frac{3}{4}\times 3\frac{3}{4}$ | $\frac{7}{8}\times 3\frac{3}{4}$ | 2 | $1\frac{1}{4}$ | $3\frac{1}{2}$ |
| $14 \times 21$ | $18\frac{3}{4}$ | 12 | $\frac{7}{8}\times 4\frac{1}{4}$ | $1 \times 4\frac{1}{4}$ | 2 | $1\frac{3}{8}$ | $3\frac{1}{2}$ |
| $15 \times 22\frac{1}{4}$ | 20 | 16 | $\frac{7}{8}\times 4\frac{1}{4}$ | $1 \times 4\frac{1}{4}$ | 2 | $1\frac{3}{8}$ | $3\frac{5}{8}$ |
| $16 \times 23\frac{1}{2}$ | $21\frac{1}{4}$ | 16 | $\frac{7}{8}\times 4\frac{1}{4}$ | $1 \times 4\frac{1}{4}$ | $2\frac{1}{4}$ | $1\frac{7}{16}$ | $3\frac{3}{4}$ |
| $18 \times 25$ | $22\frac{3}{4}$ | 16 | $1 \times 4\frac{3}{4}$ | $1\frac{1}{8}\times 4\frac{3}{4}$ | .......... | $1\frac{9}{16}$ | $3\frac{1}{2}$ |
| $20 \times 27\frac{1}{2}$ | 25 | 20 | $1 \times 5$ | $1\frac{1}{8}\times 5$ | .......... | $1\frac{11}{16}$ | $3\frac{3}{4}$ |

in Table 23 was adopted June 28, 1901, and this is suitable for pressures from 125 to 225 pounds per square inch. Flanged fittings, that is, elbows, ties, crosses, etc., are made by a number of manufacturers, and attention is drawn to the fact that the face-to-face and the face-to-axis dimensions vary with the different makes although the flange dimensions are standard. Specifications for flanges should require that the holes be drilled in the

TABLE 23.—SCHEDULE OF STANDARD FLANGES FOR EXTRA HEAVY STEEL PIPE, FITTINGS, AND VALVES.

Adopted June 28, 1901, by valve and fitting manufacturers. Suitable for pressure from 125 to 250 pounds per square inch.

| Size of pipe, inches. | Diameter of flange, inches. | Thickness of flange, inches. | Diameter of bolt circle, inches. | Number of bolts. | Diameter of bolts, inches. |
|---|---|---|---|---|---|
| 2 | 6 1/2 | 7/8 | 5 | 4 | 5/8 |
| 2 1/2 | 7 1/2 | 1 | 5 7/8 | 4 | 3/4 |
| 3 | 8 1/4 | 1 1/8 | 6 5/8 | 8 | 5/8 |
| 3 1/2 | 9 | 1 3/16 | 7 1/4 | 8 | 5/8 |
| 4 | 10 | 1 1/4 | 7 7/8 | 8 | 3/4 |
| 4 1/2 | 10 1/2 | 1 5/16 | 8 1/2 | 8 | 3/4 |
| 5 | 11 | 1 3/8 | 9 1/4 | 8 | 3/4 |
| 6 | 12 1/2 | 1 7/16 | 10 5/8 | 12 | 3/4 |
| 7 | 14 | 1 1/2 | 11 7/8 | 12 | 7/8 |
| 8 | 15 | 1 5/8 | 13 | 12 | 7/8 |
| 9 | 16 | 1 3/4 | 14 | 12 | 7/8 |
| 10 | 17 1/2 | 1 7/8 | 15 1/4 | 16 | 7/8 |
| 12 | 20 | 2 | 17 3/4 | 16 | 7/8 |
| 14 | 22 1/2 | 2 1/8 | 20 | 20 | 7/8 |
| 15 | 23 1/2 | 2 3/16 | 21 | 20 | 1 |
| 16 | 25 | 2 1/4 | 22 1/2 | 20 | 1 |
| 18 | 27 | 2 3/8 | 24 1/2 | 24 | 1 |
| 20 | 29 1/2 | 2 1/2 | 26 3/4 | 24 | 1 1/8 |
| 22 | 31 1/2 | 2 5/8 | 28 3/4 | 28 | 1 1/8 |
| 24 | 34 | 2 3/4 | 31 1/4 | 28 | 1 1/8 |

flange to straddle a vertical plane passing through the axis of the pipe. In the best work, where the flange is screwed on the pipe, the pipe and flange are carefully threaded and the pipe screwed into the flange until it projects slightly beyond its face. The inner circumference of the flange at the face of the flange is then struck with a hammer, or peaned, as it is called. The pipe and flange are then put into a lathe where both are faced off. It is very essential that the pipe and flange be faced off in the lathe in order that the face of the flange will be perpendicular to the

axis of the pipe. Another method of securing this result is to thread the flange on a mandrel so that the axis of the thread will be perpendicular to the face of the flange. Pipe over 18 inches in size cannot be threaded so the flanges are riveted on pipes over that size, or else shrunk on. In the latter case an accurately bored flange is heated and forced on the end of the pipe. Sometimes rivets are used with smaller pipe than 18 inches but more often with larger sizes. Cast-iron flanges riveted to the pipe are apt to leak as it is very difficult to make a joint that will stay tight.

The difficulty with the flange shown in Fig. 37 is the tendency to leak through the thread. This can be overcome by good workmanship, and some manufacturers have devoted a great deal of attention to making flanges of this type for high steam pressures. This type of flange is frequently put together with a copper gasket, either in the form of a flat or a corrugated ring. There are various patented packings that have also given excellent satisfaction. In the flange shown by Fig. 38, which was

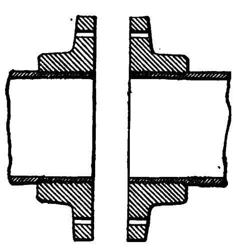


Fig. 37.

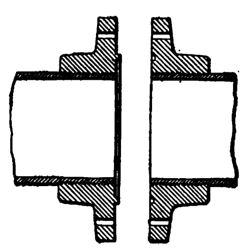


Fig. 38.

used some years ago with a good deal of success, there is a circular tongue and groove, as shown, the groove containing a ring of copper as a gasket. One objection to this flange is that in places where the piping is concentrated and the connections short, it is difficult to spring the flanges apart a sufficient amount to take out a section of pipe for repairs.

The objection to any screwed flange is the tendency, unless the workmanship be of the best, for steam to leak through the joint between the flange and the pipe. A method of overcoming this leakage is shown in Fig. 39. The sketch from which the cut was made was furnished by Mr. George I. Rockwood, Mem. Am. Soc. M. E., who designed the flange. The pipe is of steel, and

the flange is slipped over it, and the end of the pipe heated and flanged, as shown. In the original design the abutting faces of the flange are cut away so as to calk the joint. The "Walmanco" joint, made by the Walworth Manufacturing Company,

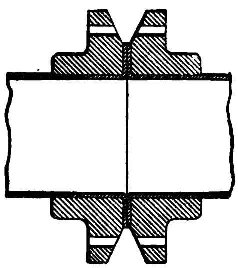


Fig. 39.

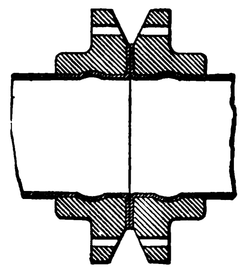


Fig. 40.

has a recess in the flange in which the pipe is expanded as shown in Fig. 40.

Cast-iron pipe with flanges cast on the ends, as shown in Fig. 41, have been used by some engineers, but the material is so treacherous in nature that its use is to be deprecated except for feed mains where the water is bad.

Steel pipe with forged-steel flanges welded on the end of the pipe, shown in Fig. 42, has been used very successfully. The

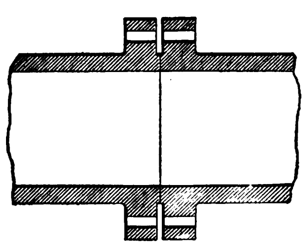


Fig. 41.

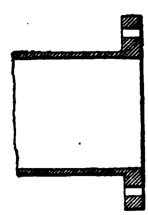


Fig. 42.

process of manufacture has been developed by the National Tube Company, which stated that in making the pipe a forged flange is bored out and forced on to the end of the pipe. It is then heated in a furnace and welded by means of a hammer. The flange is then faced and the bolt holes bored out.

The type of flange designed by Mr. Rockwood has been modified somewhat for pipe sizes 5 inches and over. The faces

of the flanges are no longer cut away on the bevel shown and the flange that is slipped on the pipe should be of rolled steel. This type of joint is used for high-grade work more than any other. In the best work it is customary to specify that the flanged part of the pipe shall be faced in a lathe on the front and back, and that after facing it shall have a thickness equal to the original thickness of the pipe. For pipe under 5 inches in diameter the joint shown in Fig. 37 is used.

**Specifications for Piping.** — When an engineer prepares accurate and complete scale drawings of the entire system of pipe fittings, etc., the written specifications can be quite brief. They should call for bids on the engineer's drawings, name the locality of the power house, give the time allowed to complete the work, and specify by name or make the following, if they are to be used, and are not marked on the drawings: Steam valves; water valves; reducing valves; back-pressure valves; steam traps; injectors; kind of packing used between the flanges; kind of pipe covering. The method of supporting pipes should be described, if it is not indicated in the drawings. The kind of fittings and pipe wanted should be clearly stated. The following extracts from a specification may be of interest:

All pipes carrying steam under boiler pressure, all pipes carrying water under the pressure of the city mains, all pipes carrying water under boiler pressure, including the blow-off piping, are to have extra heavy valves, flanges, and fittings good for 250 pounds steam pressure. In all of the piping 5 inches in diameter and over the Rockwood type of joint (sometimes called the Van Stone joint) with rolled steel flanges shall be used. The joint shall be made by slipping the rolled steel flange over the end of the pipe which shall be heated and flanged, after which the flanged part of the pipe shall be faced off in a lathe on the front, back, and on the outer rim, and after being so faced the flange shall be equal in thickness throughout to the original thickness of the pipe. In the above-mentioned piping all flanges in pipes 5 inches in size and under shall be extra heavy cast-iron flanges screwed on the pipe, flanges to be faced on a manhole after threading, or the flange shall be refaced in a lathe after it is screwed on to the end of the pipe.

All piping except that mentioned in the preceding paragraph to have standard-weight fittings, flanges, and valves except where couplings are permitted as hereinafter specified.

All pipes 2 inches in diameter and over carrying high-pressure steam or water under boiler pressure or under the pressure of the city mains, and all other pipes 3 inches in diameter and over shall have flanged fittings, and all other fittings shall be screwed fittings.

All pipes 2 inches in diameter and over carrying high-pressure steam or

water under boiler pressure and all other pipes 3 inches in diameter and over to be connected together by means of flange unions; other pipes may be joined together by standard wrought-iron couplings.

For extra heavy flanges, fittings, and valves the flange dimensions and drilling shall correspond with the 1901 Master Steam Fitters' schedule for steam pressures of from 125 to 250 pounds.

For standard-weight flanges, valves, and fittings the flange dimensions and drilling shall correspond to the Master Steam Fitters' schedule of 1894 for steam pressures up to 100 pounds.

All pipes $1\frac{1}{2}$ inches in diameter and over to have final connections to a trap, tank, heater, or any other apparatus made by means of flange unions. For pipes smaller than $1\frac{1}{2}$ inches final connections shall be made with heavy bronze unions.

All fittings in boiler-feed lines shall be extra heavy cast iron (or of cast brass made from the same pattern as extra heavy cast-iron fittings).

All flanges in high-pressure lines shall be put together with thin corrugated copper gaskets, and all flanges in low-pressure lines shall be put together with approved rubber gaskets.

Except when extra heavy steel pipe or brass pipe is specified all pipe must be full-weight steel pipe equal to the Briggs standard up to and including 10 inches, and to the Morris Tasker standard over 10 inches. Pipes carrying cold water to be galvanized.

All threads on pipes to be full and clean-cut and no other lubricant than oil and graphite shall be used in screwing up pipe. Calking of threads will not be permitted.

The boiler-feed line from the pumps to and around the heater to the boilers including the branches to be brass, iron-pipe size.

**Valves.** — It has been the practice for many years in stationary plants, particularly large ones, to use gate valves in high-pressure steam lines, while the so-called globe valve seems to have had the preference in marine work. The author's experience with gate valves for steam pressures of from 100 to 150 pounds has not been as satisfactory as with globe valves on account of an apparent greater tendency of the gate type to leakage. One objection to the use of globe valves in steam lines lies in the fact that they introduce a pocket where water might collect. By placing the valve with the stem in a horizontal position this is largely overcome, and in large valves the bodies may be dripped when necessary. It is believed that globe or angle valves might be used for sizes, 10 inches in size and under for all high-pressure steam lines, and some of these valves with extra heavy iron or steel body with special seats and disks will give excellent satisfaction. Gate valves are usually used on exhaust and low-pressure steam lines and in all water lines excepting regulating or feed valves

on boilers which should be of the globe type. Gate valves subjected to high pressure, 8 inches in diameter and over, should be provided with by-passes; when over 4 inches in size they should have rising spindles. All valves should be capable of being packed while under pressure. Two-inch valves and under usually have brass or special composition bodies with cast-iron bodies when over 2 inches in size. Some of the better valves for very high pressures have bodies of cast steel. There are few things about a power house that will cause so much trouble and annoyance as poor valves and the greatest care should be taken to make a proper selection.

**Pipe Hangers.** — Pipes should be supported in from 10- to 18-foot intervals; the smaller the pipes the more frequently they should be supported. The greatest care should be taken to hang or support pipes so they may expand freely. Bracing is necessary in some situations to stop vibration. Pipe hangers should be made of wrought iron or steel.

**Covering Pipes.** — Steam pipes should be covered for the double purpose of saving the latent heat in the steam that would otherwise be lost, to prevent steam-using machinery from being damaged by water of condensation, and in some instances, notably in buildings, for reducing the temperature of rooms in which power plants are located. With coal at $3 per ton, 10 feet of uncovered 6-inch pipe will cause an annual loss under average conditions of over $5 per year. A good covering that would reduce his loss to $1 per year will only cost about $5. From this it will be seen that it pays under most conditions to buy the best covering obtainable, and in making a selection the question of durability should be looked into fully as much as the efficiency of covering when new. Cheaper covering may be used on heating and low-pressure lines when the greater heat transmitted by the cheaper covering will not cause a loss. In underground power plants, where artificial ventilation is necessary to reduce the temperatures to an endurable degree, the best covering for all hot pipes is worth its cost.

Some extracts from a specification for pipe covering follow.

All hot pipes furnished under this contract, also the branch feed pipes on the boilers as far as the stop valves, shall be covered with magnesia sectional covering of one or more thicknesses as specified. Drip pipes in trenches are not to be covered. All sectional covering shall be neatly banded with black iron bands.

All high-pressure steam lines shall have a double nonconducting covering equal to double the standard thickness as given in the catalogues of the —— Co., or the —— Co., and all covering shall be furnished and applied by one of the firms mentioned, or by any other manufacturer satisfactory to the engineer. Where double thicknesses of covering are applied they shall be laid so as to break joints as far as possible.

All fittings and valves excepting valve bonnets shall be covered with block magnesia of the same thickness as covering on adjacent pipes and finished in a hard plaster. The flanges shall be covered with plastic magnesia 1 inch thick and finished in a hard plaster.

All sectional covering shall be round and smooth and all ends butt evenly and tightly together. No damaged or broken sections shall be used. When covering is formed from blocks care shall be taken to see that the blocks are properly applied and securely wired, with joints closed with plastic magnesia.

The steam separators, the reheaters on the compound engine, and all steam piping furnished by the engine builder, the grease separator, feed-water heater, the return drip and blow-off tanks, and the heads of the boiler drums shall be covered with 2-inch magnesia blocks wired on and then finished in a hard plaster.

All nonconducting material on exposed pipes shall be covered with resin-sized paper over which there shall be an outer covering of 10-ounce canvas sewed on and painted. All block or plastic covering shall be covered with 8-ounce canvas neatly cut and fitted, pasted on and painted.

Contractor shall cover the steam cylinders and valve chests of all pumps with magnesia blocks 2 inches thick, put on so as to be easily removable.

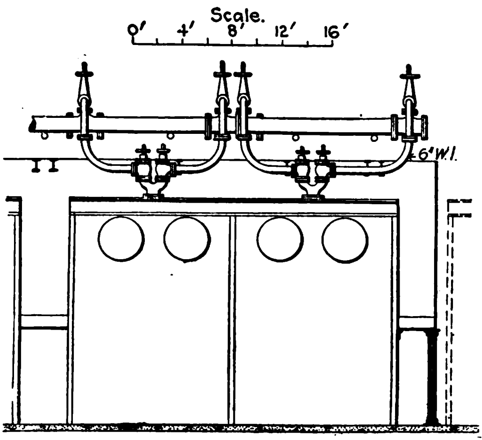
All so-called magnesia covering shall contain not less than 85 per cent carbonate of magnesia.

Paint all nonconducting material exposed to view with two coats of cold-water paint, excepting within five feet of the floor, where two coats of lead-and-oil paint of color selected by the engineer shall be used.

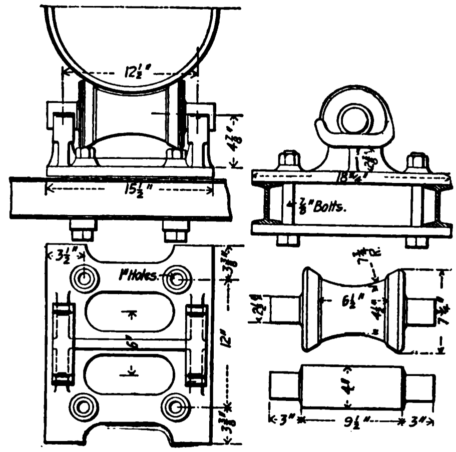
Cheaper forms of covering than 85 per cent carbonate of magnesia are the so-called air-cell coverings made by several manufacturers, in which one layer of asbestos paper fluted to form the air cells is wrapped over another layer until a sufficient thickness is obtained. When it is desired to economize this or other cheaper coverings may be used on low-pressure work, and the outer canvas jacketing and painting may be omitted and all plastic or block magnesia may be finished in a hard plaster.

**Grease Separator.** — In noncondensing plants where the exhaust steam is used for heating or any manufacturing purpose a grease separator should be used. This may be the small separator, usually of cast iron, which forms a slight enlargement in the exhaust pipe, with various forms of baffle plates that are intended to separate the grease and allow it to flow into a chamber in the bottom of the separator, from which it is drawn off by a trap. A more efficient type is that in which a large steel tank, about 4 feet in diameter and 8 feet long for a 14-inch main, is

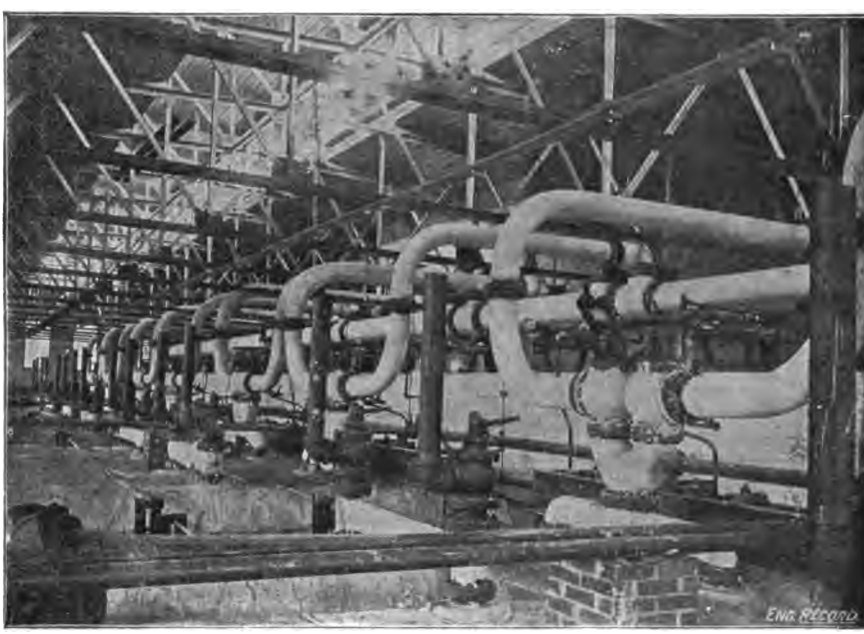
[illegible]




ELEVATION OF BOILER PIPING.




DETAIL OF PIPE SUPPORT.




METHOD OF CONNECTING DUPLICATE MAINS TO BOILERS.

Fig. 43. Steam-piping Details at Great Northern Paper Co. (Sheaff and Jaasted, Engineers.)

used, the tank being placed horizontally. The steam enters the top at one end and passes out at the other end of the tank, also at the top. Various forms of baffles to catch the grease are placed within the tank. The large sectional area of the separator gives the particles of grease and condensed steam a chance to fall to the bottom and collect there, from which they are drawn off by a trap. Separators of this type, and there are several on the market, are highly efficient, and if properly proportioned and used will remove practically all of the oil from exhaust steam, so that the condensation may be safely returned to the boilers. This separator, however, will not remove the oil sufficiently to overcome a cloudy appearance to the condensation of exhaust steam, due apparently to the fact that a certain part of the oil is volatilized and passes through the system in volatile form to be condensed on such cooling surfaces as it may come in contact with.

**Steam Separators.** — Steam separators are placed in steam pipes for the purpose of removing any condensation that may form or find its way into the pipe. They are made either in the form of a casting or a receptacle made of sheet steel with suitable inlet and outlet nozzles, and of sufficient volume to allow the steam to be so reduced in velocity as to permit the water to collect in the separator from which it may be drawn off by a trap. The receiver separator is usually placed close to a steam-engine cylinder, sometimes on top of the throttle valve, and its volume is so large as to furnish a reservoir of steam at the engine and do away with the intermittent flow of steam that occurs in the steam pipe, due to the opening and closing of the engine valves. This intermittent flow of steam has proved to be very objectionable in power plants in buildings, by causing vibration which has been overcome by the installation of a large receiver separator of this type. They are usually made with a volume two or three times the volume of the engine cylinder. Separators are absolutely necessary for steam engines with a positive stroke when there is any chance of water being present in the steam, to prevent the engine from being wrecked.

**Steam Traps.** — There are many kinds and makes of steam traps on the market, and while many have given excellent satisfaction one should consider that it is only a question of time when a trap will begin to leak and be a source of expense. Traps are of the pot, float, or expansion type. The pot traps are those

having a pot in the interior that when filled by the entering condensation operates the valve, allowing it to discharge in an intermittent manner. Float traps, as the name implies, contain a float inside of a receiving chamber controlling the discharge valve, opening when the condensation in the trap reaches a certain level. Expansion traps contain some expanding device that closes the discharge valve when exposed to the higher temperature of steam and opens when in contact with cooler water of condensation. In selecting traps care should be taken to see that parts likely to wear, such as valves and seats, may be renewed easily and replaced at slight expense. Traps should be connected to piping by means of brass unions or flanges so they may be easily disconnected for repairs, and they should be provided with a by-pass with the necessary valves, usually three, so the trap may be used or not as desired. Globe valves should be used in trap connections. Traps will not discharge naturally against a greater pressure than the pressure of steam entering them, a fact that is sometimes overlooked in steam-plant design.

**Engine-oiling Systems.** — There are two systems usually used for supplying oil to engines; one in which sight feed oil cups are placed on the various bearings, and the combination of these cups and a central oiling system where oil is pumped into elevated tanks and is thence conveyed by gravity through a system of pipes to the bearings, from which it is collected and run also by gravity to an oil filter located at some lower level than the engines. The filtered oil is again pumped to the overhead tank and is thus used over and over again, thereby effecting a considerable saving in the amount of oil used. Oil feeds used in this system are usually fed either from the central oiling system or from the cup itself which may be filled by hand. With the central oiling system a much more liberal supply of oil may be supplied to a bearing, thus producing better lubrication and to some extent less frictional loss. Cylinder lubrication may be obtained from the sight feed gravity oiler in which the pressure of water in a vertical pipe about 3 feet long, extending from the steam pipe to the lubricators, is utilized to force the oil into the steam-supply pipe. A much better device is the more modern force-fed lubricator which consists of an oil reservoir fitted with a pump attached to some moving part of the engine so as to supply a definite quantity of oil with each stroke of the engine.

These pumps are particularly valuable in lubricating pumps when the speed is variable.

A specification for an oiling system for an office building plant is given in the chapter on " Engine Specifications " in the specifications for a Corliss engine.

# CHAPTER IX.

## CONDENSERS AND PUMPS.

THE purpose of attaching a condenser to a steam engine is to remove part of the pressure of the atmosphere, by creating a partial vacuum, from one side of the piston, and thus increasing the effective pressure acting upon the other side. If an engine is running without a condenser, with a mean effective pressure of 40 pounds, and a condenser is added removing 12 pounds from the back pressure caused by the pressure of the atmosphere, the mean effective pressure would be increased, theoretically, to 52 pounds, and the power would be correspondingly increased. If, however, the work done remains the same, the increased mean effective pressure, 52 pounds, can be reduced to 40 pounds by cutting off the steam earlier in the stroke and thus save steam nearly in the proportion that the cut-off is reduced. A condenser can, therefore, either increase the capacity of an engine, or it will reduce its steam consumption per horse-power if the load and other conditions remain the same. If an engine is operated noncondensing with the most economical load for it and a condenser be fitted to the engine, this same load will no longer be the most economical one, although the steam consumption per horse-power per hour will probably be considerably less with the condenser than without it. A condensing engine should have a larger cylinder than a noncondensing engine to do the same work, if both are to run with the lowest possible steam consumption per horse-power. The method of determining proper cylinder dimensions for engines running condensing and noncondensing was given in Chapter IV.

The attachment of a condenser to a turbine has the same effect that it has upon a steam engine in that it increases the work that may be obtained from the same quantity of steam, or, with a condenser, less steam will be required to do the same work. There is opportunity for a condenser to save more with a turbine than with a steam engine, as it is possible to operate turbines with as

great a number of expansions of steam as possible, whereas, in a steam engine, the number of expansions is limited by cylinder condensation and the high cost of the large cylinder sizes necessary for a high number of expansions.

A condenser provides means for bringing cool water into contact with exhaust steam or passing it through thin tubes around which the exhaust steam passes, so that the latter is condensed. A vacuum is formed in the condenser by reason of this condensation, and thus part of the pressure of the atmosphere is removed. Because of the small volume occupied by the condensed steam relative to its volume as steam at the same temperature, condensed steam or water may be removed from the condenser by a comparatively small pump using a small amount of power to operate it. This pump also removes such air as may be in the exhaust steam, hence it is usually called an air pump, although it serves to remove the water as well.

**Saving Due to Condensers.** — Generally speaking, a condensing engine will use from 75 to 80 per cent of the steam required by a noncondensing engine. Power, however, is required to drive the air pump, and this saving is, therefore, not a net gain. The steam required to drive an independent steam-driven air pump and circulation pump is from 6 to 10 per cent of that used by the main engine, depending on the size of the latter. This use of steam for driving the air pump need not necessitate a loss, if the exhaust steam from the pump is used to warm the feed water before the latter is delivered to the boilers. If the feed water is drawn from the air-pump discharge or if it is fresh water heated in a feed-water heater placed in the exhaust pipe between the engine and condenser, 110° F. is about as high a feed temperature as can be obtained on account of the vacuum in the condenser and the correspondingly low temperature of the exhaust steam. It is, therefore, advantageous to use the steam exhausted by the condenser pumps for feed-water heating in an auxiliary heater, so-called because it receives the exhaust steam of the auxiliaries of the plant, such as the condenser and boiler-feed pumps. This heater is sometimes called a secondary heater when a primary heater is placed in the exhaust pipe of the engine. By means of the auxiliary heater the feed water may be raised from 110 to about 180 degrees in a large, well-proportioned plant, and to a greater extent in smaller plants, as the steam required for the

condenser pumps and boiler-feed pump becomes a greater proportion of that used by the main engine.

With a noncondensing engine the feed water may be heated by the exhaust steam in a feed-water heater to within about 10 degrees of the temperature of the steam or to about 200 degrees with steam at atmospheric pressure; consequently, with condensing engines, the saving in coal used by the main engine due to a condenser over the use of the same engine running noncondensing is partly counterbalanced by the higher feed temperature that may be secured when running noncondensing. With feed water taken from the condenser discharge at a temperature of 110 degrees, there would be a saving, with a steam pressure of 100 pounds, of about 9 per cent if the feed water could be further raised in temperature to about 205 degrees, as it probably could be if the engine was run noncondensing. However, by using the exhaust steam from the air and boiler-feed pumps of a condensing plant to warm the feed water taken from the air-pump discharge, there will be such a small difference between the final feed temperatures in the two types of plants that the slight difference can be neglected in considering the saving due to the use of a condenser. It is impossible, and perhaps unnecessary, to figure the exact saving a condenser will produce, but, generally speaking, it may be taken at about 20 per cent, hence plants are almost always fitted with these auxiliaries when condensing water is to be had, and the exhaust steam is not needed for heating or for some manufacturing purpose. When an abundant supply of condensing water is not available, steam-plant owners often go to considerable expense for artificial cooling devices, so great is the economy in using condensers.

**Condensers.** — With turbines the saving due to condensers is more marked than it is with a steam engine, the amount depending upon the size of the turbine and the initial steam pressure. With turbines a very large part of the work done is due to the expansion of steam from atmospheric pressure down to the pressure of high vacuum, and there is a very marked change in the steam consumption due to increasing the vacuum from 27 to 28 inches, the latter figure being common in turbine practice. The accompanying table (24), gives the steam consumption guaranteed by the manufacturer of a 1000-kw. turbine operating with steam at 150 pounds gauge pressure under different conditions of load and vacuum.

TABLE 24. — STEAM CONSUMPTION.

| Vacuum. | $\frac{1}{2}$ load. | $\frac{3}{4}$ load. | Full load. | $1\frac{1}{4}$ load. |
|---|---|---|---|---|
| 27 | 23.6 | 21.0 | 19.8 | 20.75 |
| 28 | 21.9 | 19.7 | 18.7 | 19.0 |
| 29 | 20.3 | 18.4 | 17.5 | 17.7 |

It will be noticed that at full load there is a reduction in the steam consumption of over 11 per cent due to increasing the vacuum from 27 inches to 29 inches. This, however, is not all net gain as a greater expenditure of fuel is required to maintain the higher vacuum.

**Type of Condenser.** — Condensers may be divided into three general types, known as the jet, the surface, and the siphon or barometric condensers. Up to the advent of the turbine, condensers were comparatively simple, the jet and surface condensers requiring an air pump either directly driven from the engine or driven by steam cylinders forming part of their equipment. Even with the barometric condensers a form of air pump known as a dry-vacuum pump was sometimes used with large engines. The direct-connected air pump is now seldom if ever used with engines for power purposes, the satisfactory manner in which the independent condensers have been perfected and their superiority over the direct-driven air pump having brought this about. The development of the steam turbine, requiring as it does a very high vacuum and the immense size of turbines, several units of 20,000 kw. having been manufactured, requiring a supply of from 40,000 to 50,000 gallons of cooling water per minute, have completely revolutionized condenser practice, and all kinds of combinations have been brought out.

The jet condenser in its simplest form usually consists of a pear-shaped chamber at the top of which the exhaust steam enters, while at one side is a connection for the injection or condensing water. The bottom of the chamber has a contracted neck connecting with an air pump which is sometimes very similar to the ordinary direct-acting steam pump, or it may be an air pump of special design. A cross section of the Blake jet condenser with horizontal double-acting air pump is shown in Fig. 44.

The surface condenser consists of a shell, usually of cast iron, containing a large number of brass or copper pipes through which the condensing water is circulated, the exhaust steam being admitted to the shell in the space surrounding the pipes. The steam coming in contact with the cooled pipes is condensed, and the condenser is so connected with the air pump that the con-

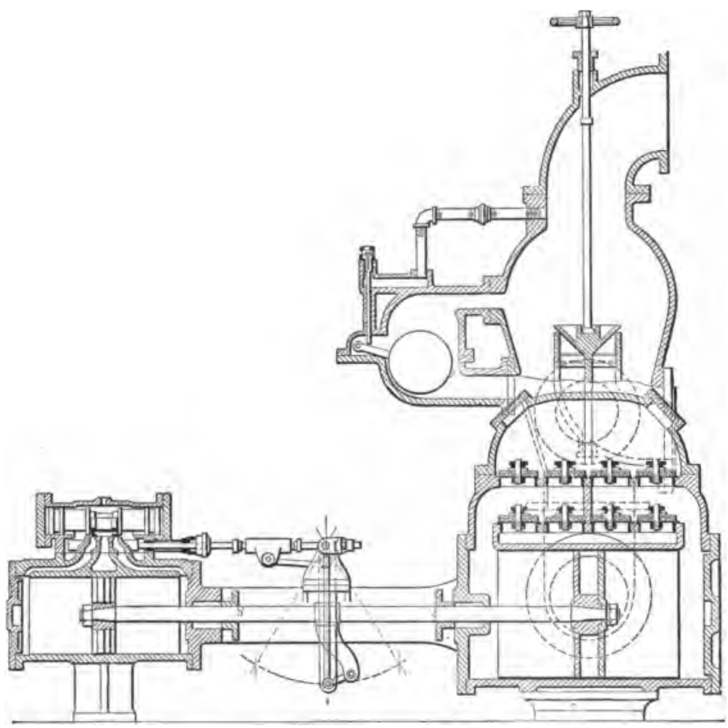


Fig. 44. Section of Blake Jet Condenser.

densed steam flows by gravity to the air pump, by which it is removed to maintain the vacuum. If the cooling water is under pressure no pump is necessary to circulate it through the condenser. If not, a circulating pump is required and this may be driven by the same steam cylinder that operates the air pump. It is usually the practice in small condensers to place the air and circulating pump and the steam cylinder operating them in line, tandem, beneath the condenser and on a base supporting the whole. An arrangement of this kind showing the Wheeler

method of mounting a surface condenser on the cylinders of a Knowles' air and circulating pump is shown in Fig. 45. Circulating water is sometimes supplied by centrifugal pumps driven by an engine or electric motor. As the condensed steam does not come in contact with the circulating water in the surface condenser, this type can be used when the circulating water is of such a character that it should not be fed to the boilers. The condensed steam is used over again until it becomes so impregnated with cylinder oil from the engine that it has to be replaced

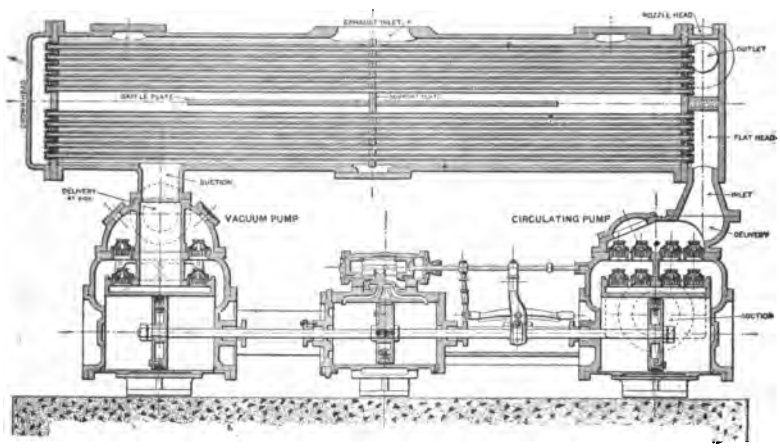



Fig. 45. Knowles' Air and Circulating Pump.

by fresh water. The presence of cylinder oil in feed water causes trouble in steam boilers, and it is the fear of this trouble from oil that prevents the wider adoption of the surface condenser in power plants. However, by providing proper filters of sand, cloth, sponges, excelsior, or similar material for the feed water and properly looking after them, there is no reason why the surface condenser cannot be used with success. In certain rivers which are sour and contaminated with sewage the surface condenser has proved to be very costly, due to the rapid deterioration of the tubes.

Figure 46 shows in plan an elevation of a Worthington surface-condensing equipment as applied to a 3000-kw. Curtis turbine. Cooling water is supplied by an engine-driven centrifugal pump, and a motor-driven centrifugal pump draws the condensed steam

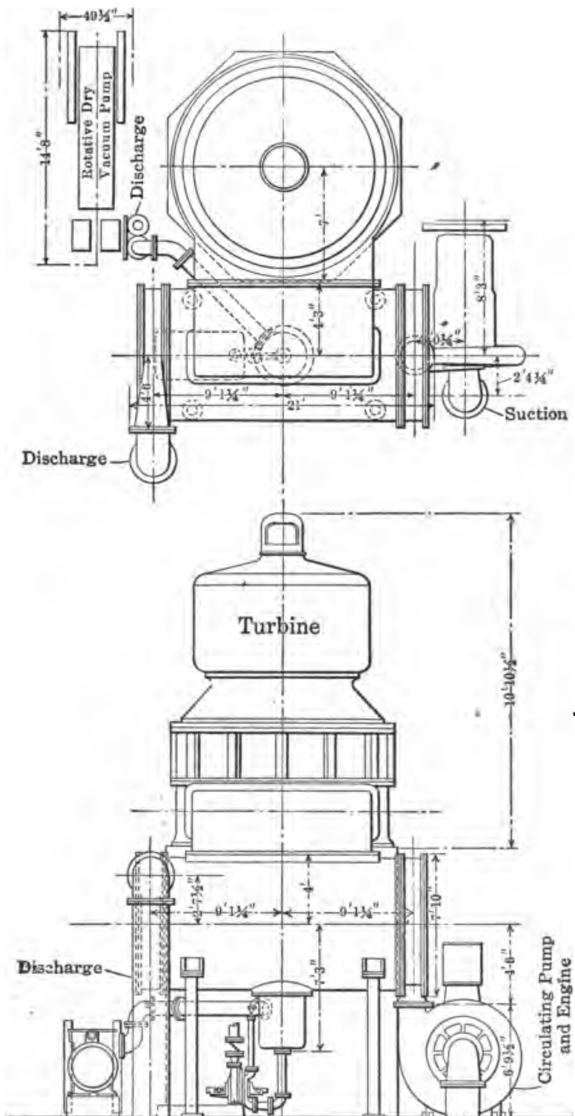



Fig. 46. Plan and Elevation of a Worthington Surface Condensing Equipment as applied to a 3000-kw. Curtis Turbine.

from the hot well, while an engine-driven, dry-vacuum pump is also connected to the hot well.

The Bulkley siphon condenser, which in a general way is similar to others of this type, is shown in Fig. 47. The steam from the engine is led in a pipe to the top of the condenser, which is elevated sufficiently to be placed about 34 feet above the surface of a hot well into which the condenser discharges. The injection water enters at the side and mingles with the steam at the lower edge of the cone shown. By contracting the neck of the condenser below the cone, sufficient velocity is given the water in falling to the hot well to maintain a siphon-like action that draws the air and noncondensable vapors from the exhaust pipe, and causes a vacuum to exist in it. The injection water can be supplied by a pump of either the steam-driven or the centrifugal type. If the injection water can be had under sufficient pressure, the pump is not necessary. This type of condenser will lift water from a source of supply, such as a tank or reservoir, through a height of 18 feet or less, but with this arrangement the siphon must be started. This can be done by running a horizontal pipe from the reservoir or tank across to a tee in the vertical discharge pipe of the condenser. Water flowing through this and down the discharge will gradually exhaust the air from the upper part of the discharge pipe until sufficient vacuum is formed to draw the water up to the condenser and start the water flowing through it. When this is done a valve in the cross connection or starting pipe is closed.

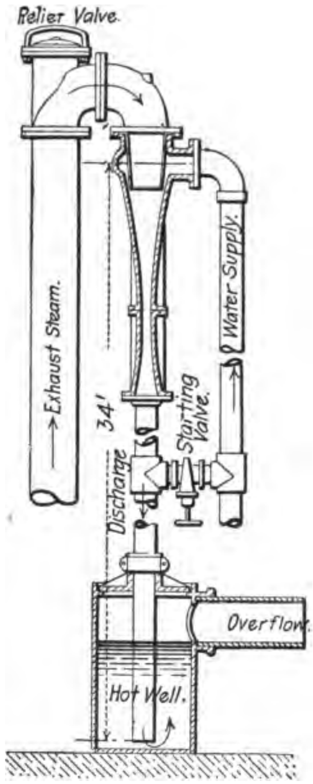


Fig. 47. Siphon Condenser.

Figure 48 shows an elevation of the Worthington elevated counter-current jet condenser which is a modification of the

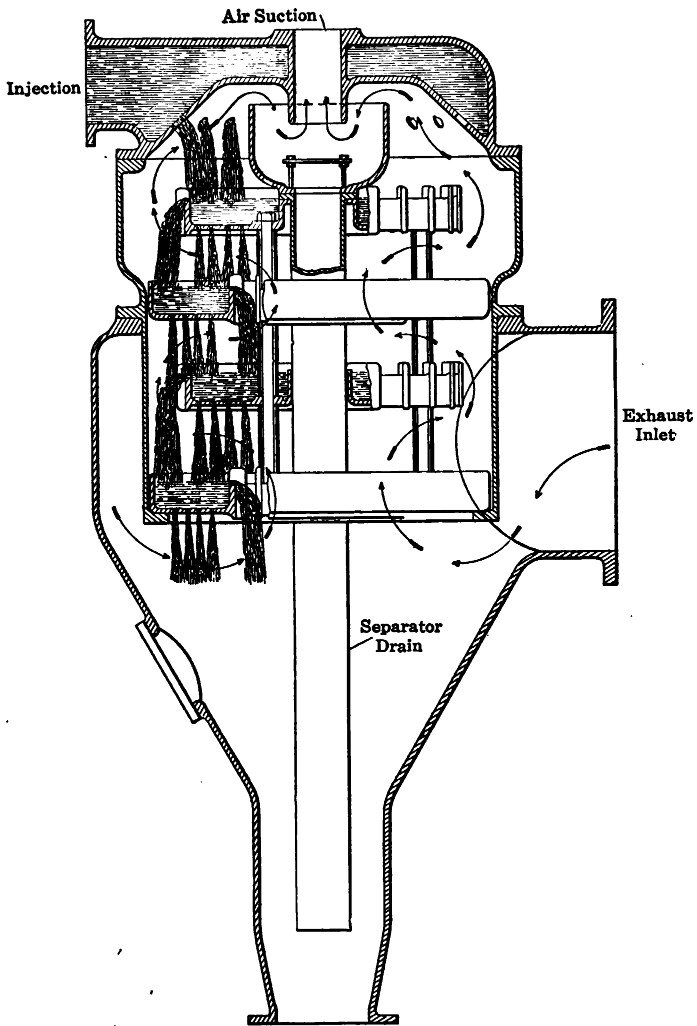



Fig. 48. Elevation of the Worthington Elevated Counter-current Jet Condenser.

siphon type. The condensing cone is considerably larger than in other siphon condensers, the idea being that the steam and water are more thoroughly mixed, thus using less water. The neck of the condenser is not contracted, as it is in the ordinary siphon condenser, to give the high velocity to the descending

water necessary to suck the air down into the discharge or tail pipe. The contracted neck was avoided to give the water an unrestricted fall so that it could not by any chance back up in the condenser and run over into the exhaust pipe. To assist in removing the air from the condenser cone the top outlet is connected to a dry-vacuum pump, which constantly removes air from this pipe and very considerably increases the vacuum over what would be obtained without it. The entering cooling water falls down through the successive trays in a fine spray so as to hold it in a finely divided state in contact with the steam as long as possible.

While the above describes briefly the various types of condensers in their simplest forms they have been elaborated to a considerable extent. With turbines the power required to drive the various air and circulating pumps is so great that Corliss engines of the highest economy are frequently used for this purpose. The condenser for the 12,000 kw. with the Curtis vertical turbine in the Commonwealth Edison Company's station in Chicago is located beneath the turbine and forms a base for it. The impeller of the circulating pump is mounted on the end of the shaft of the Corliss engine driving it, and the air pump is placed in the rear of the engine cylinder, and is operated by a tail rod passing through the back head of the steam cylinder. In other cases the air pump is operated in a similar manner with a separate centrifugal pump furnishing the circulating water and driven at a higher rotative speed, either by a high-speed steam engine or by an electric motor or a steam turbine. A typical case of this kind was shown in Fig. 46.

The jet type of condenser is less expensive than the surface and very excellent results have been obtained by connecting the discharge of the condenser directly into the suction of an engine- or motor-driven centrifugal pump with a separate dry-vacuum pump connected to the condenser head for removing the air.

As to the relative advantages of different types of condensers the surface type is the most efficient, and it is possible to secure a higher vacuum with this type than with the others, the conditions being equally favorable. This is largely due to the fact that the cooling water does not come into direct contact with the steam, hence the air-removing apparatus does not have to remove the vapors from the cooling water. Furthermore they

can, generally speaking, be located more favorably than other types as they can be connected directly to the discharge passage from certain types of turbines.

**Location of Condensers.** — It is important to locate a condenser of the jet or surface type on a lower level than the engine or turbine, so that the pipe connecting the engine and condenser will be either perfectly horizontal or pitch slightly toward the condenser. This is necessary to prevent the existence of a pocket where water can collect in the pipe line. As this pipe is under a vacuum, water due to condensation that collects in it cannot be drawn off by a trap. A vacuum trap might be used but it is advisable not to use one if possible. If a pocket does exist there is chance, in the event of broken vacuum, of this water getting back into the engine cylinder and wrecking it. One occasionally sees a condenser located on the floor with an engine. When this is done the exhaust pipe from the engine drops from the bottom of the cylinder, then runs horizontally, then rises to the condenser, thus forming a pocket, to which objection has been raised; this arrangement should be avoided.

There being a vacuum in the condenser, the injection water can be lifted from a reservoir or pond at a lower level. It is not advisable to lift water through a greater head than 20 feet, including the actual vertical distance the water is raised and also the friction of the water in the injection or suction pipe. A condenser is sometimes located in a pit considerably below the engine or turbine in order that the lift may not be too great. As stated in the first chapter of this series, the availability of a supply of condensing water often determines the location of the power house. If a river or reservoir is near by and it is not desirable to locate the power house close to the river, water can be led to a well close to the condenser either in an open trench or in a tunnel or conduit and the injection pipe run to the well. This would overcome the use of a long injection pipe. If the injection pipe is run underground it should not be covered over with earth until after the condenser is started and the pipe tested for air leaks. One of the most frequent causes of trouble with a condenser is a leaky suction pipe, hence the greatest care should be used to make sure that it is perfectly tight. Various methods of connecting condensers with engines were given in the chapter on "Steam Piping."

**Water Necessary for Condensers.** — A considerable amount of water must be available if a condenser be used, and to calculate this, one must first find the weight of water necessary to condense each pound of exhaust steam and then determine the amount necessary to condense all the steam exhausted by the engine or turbine. The former is found by dividing the rise in temperature that takes place in water used to condense the steam into the heat contained in one pound of steam at the pressure at which the exhaust valve in the engine begins to open less the heat in one pound of water at the temperature of the air-pump discharge. Expressed algebraically, this equation is: $W = (H - h) \div (T - t)$; in which $H$ is the total heat in steam at the terminal pressure, $h$ is the heat in water at the temperature of the air-pump discharge, $T$ is the temperature of the discharge condensing water, and $t$ is the temperature of the entering condensing water. The value of $H$ for different pressures and temperatures may be found in tables giving the properties of saturated steam, and may usually be taken at 1150. Taking average values for a surface condenser with $h$ at 120, $T$ at 110, and $t$ at 70, then $W$ will be found to have a value of about 26 pounds per hour. If a jet condenser is used the condensed steam and air-pump discharge would have the same temperature. When the same water is used over and over again to condense with, as is done with cooling towers or cooling reservoirs, the temperature of the condensing water is quite high when it enters the condenser, so that each pound can absorb a comparatively small amount of heat, hence a correspondingly greater volume of condensing water is required.

With turbines under 28 inches of vacuum and a cooling water temperature of 70 degrees in summer, average practice is to so design the condensing system that the circulating water when leaving the condenser will be within at least 15 degrees of the temperature of the steam entering the condenser and an increase in the temperature of the cooling water of at least 15 degrees. As the total heat in steam at 1 pound absolute pressure is 1145, and as $k$ would have a value of 101, $T$ would have a value of 86 and $t$ a value of 7.0 Hence the amount of cooling water required would be a little over 65 pounds per pound of steam condensed. If the amount of cooling water is increased the amount of surface required in a surface condenser could be decreased, but at an

expense of the greater amount of power required to drive the circulating pump. Again if the volume of water is decreased the discharge water would be of a higher temperature, hence the difference between the mean water temperature and the steam would be less and a greater surface would be required in the condenser.

In estimating the quantity of injection water necessary, due consideration should be given the amount of steam exhausted by the engine or turbine during maximum load. The author believes in being very liberal in selecting condensers, for a good vacuum, particularly with engines operating with low mean effective pressures, is conducive to high economy. With turbines a good vacuum is imperative. Some engineers hold that a high vacuum with engines is a mistake, one of the reasons being that with it the temperature of the condenser discharge is lower and consequently the feed water will not be so hot. There is, however, a loss due to reducing the mean effective pressure acting on the engine piston that considerably more than offsets such a gain. By a decrease in the vacuum from 26 to 24 inches it would be impossible to increase the feed-water temperature more than 15 degrees, and this would mean a saving of a little more than 1 per cent. This decrease in vacuum in a compound engine, with 125 pounds steam pressure, most economically loaded would effect, theoretically, an increase in the steam used of about 5 or 6 per cent, and with a simple engine about 2.5 per cent. It will be noticed on investigating duty trials of large pumping engines, where high duty is desired, that a very high vacuum is sought by the builder. The effect of condensers upon steam turbines is discussed briefly in the chapter on "Turbines."

**Sources of Water Supply for Condensers.** — Water for condensing purposes may be obtained from rivers, in which event the water goes to waste after it is discharged from the condensers, or from ponds or artificial reservoirs. When drawn from ponds or artificial reservoirs the source of supply has to have sufficient volume and surface so that it will be cooled naturally by the air. If the reservoir is too small for this it must be cooled artificially by some method that exposes the water sufficiently to the air to cool it, as in the cooling tower. Reservoirs where the cooling is done naturally require surface sufficiently large to allow enough water to come in contact with the air to remove the necessary amount of heat from it, and they should also have sufficient vol-

ume, if they are used continuously, so that the water can remain in the reservoir long enough to be cooled before being used again. Cooling reservoirs of this kind are rendered much more efficient by dividing them by partition walls so that the water is compelled to travel some distance in passing from the condenser discharge to the intake where it is drawn from the reservoir to return to the condenser. This is to prevent the discharge from the condenser from immediately entering the intake, or short-circuiting, before it has time to cool.

There is very little reliable information on the surface and volume required in cooling reservoirs. Thomas Box, in his work on "Heat," says that when an engine works day and night the depth of the reservoir is unimportant, and that 210 square feet of surface is required per horse-power. When an engine runs only 12 hours per day the surface may be reduced to 105 square feet per horse-power, but in the latter case the depth of the reservoir must be such as to give 300 cubic feet per horse-power. Box assumes the use of one cubic foot or $62\frac{1}{2}$ pounds of water evaporated into steam per horse-power. These data are based on experiments when the water was reduced in temperature from 122° to 82° F., the air being at a temperature of 52 degrees and the humidity 85 per cent.

**Cooling Towers.** — These appliances provide means for artificially cooling condensing water, and their successful development has made it possible to obtain the benefit of a condenser in many plants where condensers would be out of the question without them. Usually they consist of large cylinders of sheet steel open at the top and enclosing either mats or tiles or some similar substance presenting a large surface over which the hot water from the condensers is allowed to trickle downward, from distributing pipes at the top. The water falls into a reservoir at the bottom, from which it is returned to the condenser. An opening or openings in the side of the tower allow the air to enter and pass up through it, thus cooling the water. Usually a fan driven by an electric motor or small steam engine is used to stimulate this current of air. A cooling tower can be located almost anywhere outside of a power house, on the ground or on the roof. There is a loss of circulating water attending their use of from 10 to 15 per cent of that passing through them, owing to the evaporation that occurs. This, however, is more than made up, with jet con-

densers, by the discharge from the air pump of the condensed steam. Where cooling towers are located at a higher elevation than the engine, the condensing water must be pumped from the condenser to the tower, but the power required to do this is,

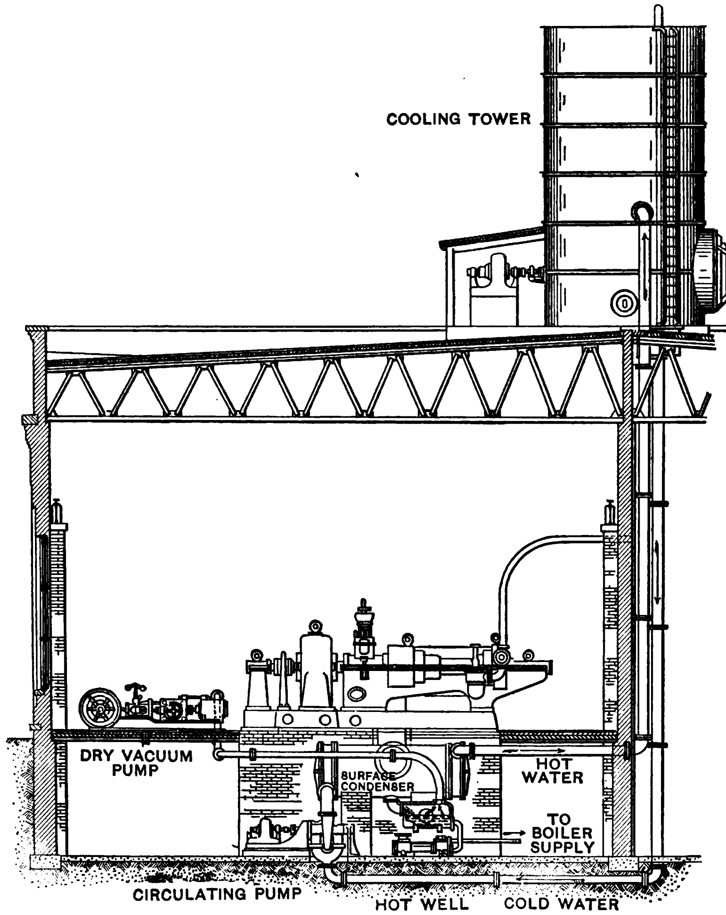



Fig. 49. Alberger Cooling Tower.

with surface condensers, partly counterbalanced by the fall of water from the reservoir at the base of the tower to the condenser. The column of water in the condenser discharge pipe for the height of the tower itself is, of course, unbalanced. A view of an Alberger cooling tower as applied to a surface condenser for a turbine is shown in Fig. 49.

**Specifications for Condensers.** — Generally in purchasing jet condensers for an engine an engineer states the horse-power and type of engine for which a condenser is wanted, but it is better, perhaps, if the engineer is sure it will be correct, to give the number of pounds of steam that must be condensed in a given time, and the vacuum that is desired. This is usually 26 inches in engines. Specifications should also call for a blue-print or drawing showing the condenser the builder intends to supply, and on this blue-print should be given the outside dimensions of the condenser, the size of the steam cylinder, the diameter and location of steam- and exhaust-pipe openings, the injection or suction pipe, and the discharge pipe. The delivery and erection of the condenser should be provided for, if the builder is to deliver and install it.

A specification for a surface condenser with combined air and circulating pumps should ask for a print of the apparatus, the size of the air, water, and steam cylinders, the size of the steam and exhaust pipes, etc., also the square feet of heating surface in the condenser proper.

When it comes to buying a condenser for a turbine plant the advice of reliable and experienced firms manufacturing condensers for this class of work should be obtained. In this case the engineer had better let the contractors submit their own plans and specifications.

**Guarantees.** — Sometimes a condenser manufacturer is required to guarantee its efficiency, but this is not often done. The form of guarantee that is fairest to the manufacturer would be, perhaps, to ask that the apparatus with an air-pump piston speed not exceeding a specified amount, should condense a given amount of steam delivered to it at a certain pressure, and maintain the required vacuum, with a given temperature of cooling water. The location at which the vacuum is to be obtained should be stated, for frequently there is considerable drop in pressure between the exhaust pipe next to the engine and the condenser, owing to a too long or too small exhaust pipe. The vacuum ought to be measured close to the condenser, for the maker of the latter may not be responsible for the selection of the exhaust pipe. He may not be responsible for the size and run of the injection pipe leading cold water to the condenser, or for the lift of water in it, and as these very materially affect the efficiency

of the condenser, the manufacturer should have the opportunity, if a guarantee is made, to approve the details, size, and arrangement of the injection pipes. These remarks apply also to jet and siphon condensers. With a surface condenser and combined air and circulating pumps, the only guarantee that should be asked is maintaining a specified vacuum with a piston speed of the air pump not exceeding a specified amount and a specified temperature of cooling water. The manufacturer would have to satisfy himself with the arrangement of the water piping that it was proposed to use. If a complete cooling tower and condensing outfit was to be supplied the guarantee should state the vacuum that is to be obtained with a given temperature of outside air, and when that air contains a certain percentage of moisture.

**Boiler-feed Pumps.** — Boiler-feed pumps may be of two general types, the direct-acting steam pump and some multi-stage form of centrifugal pump, the latter driven either by an electrical motor or by a steam turbine. Centrifugal pumps are used quite successfully in very large power stations and they are generally turbine-driven. The direct-acting boiler-feed pump may be the ordinary piston-pattern pump, the outside center-packed-plunger pump or the outside end-packed-plunger pump. Plunger pumps may be easily packed without opening up the pump chambers. The piston pump is the cheapest and the end-packed pump the most expensive and likewise the best for high pressures and large pumps. Again, a pump may have valves in the pump chamber or the pump may have the more accessible pot valves. The pot type of valves are much more accessible.

The center-packed-plunger pump has four separate water chambers for the purpose of reducing the expense of replacement should damage occur. The piston pump requires less space and the outside-end-packed the most space. In the pot valve end-packed water end there are four cylinders, two on each side, cast together with a diaphragm in the center. Four single-acting plungers work in the ends of these through stuffing boxes, the plungers on each side being connected together by crossheads and tie-rods; thus no piston rod enters the water cylinders. The chambers are cast separate from water cylinders and the suctions and discharge valves may be gotten at without disturbing any part of the water end. Sectional views of the end-packed pot-valve pump are shown in Fig. 50.

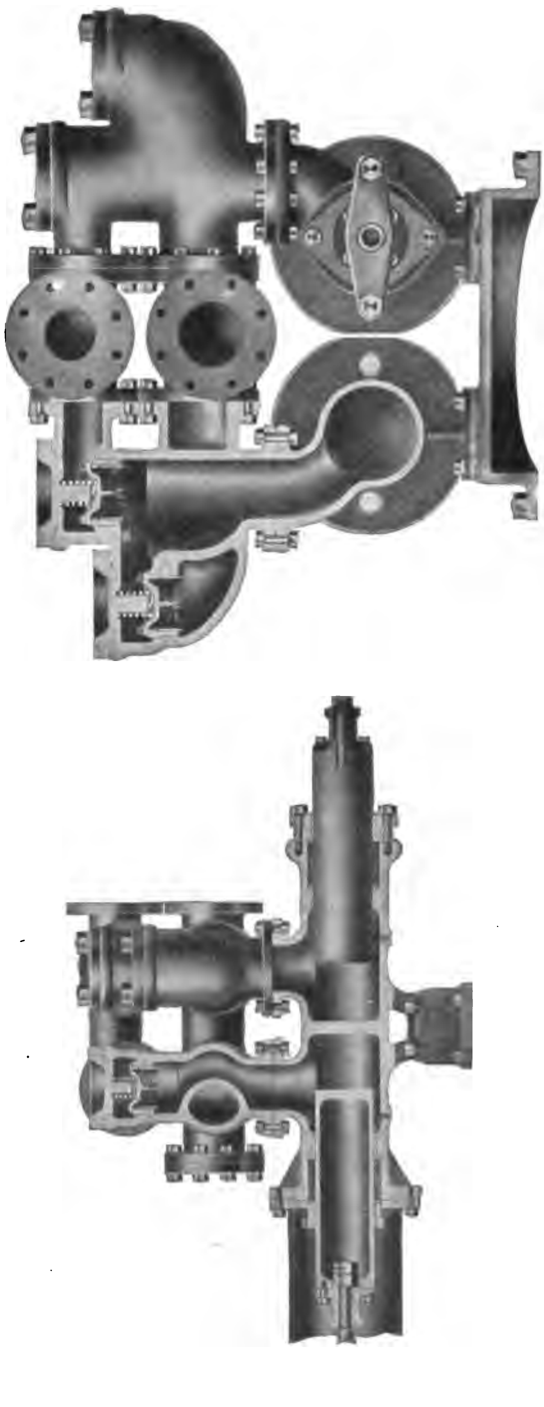


Fig. 50. Sectional Views of the End-packed Pot-valve Pump.

For building work for boiler plants of 500 horse-power and under the piston pump is usually used, and in order that it will pump hot water the pump should be brass fitted, that is, provided with brass valves, pistons, and bronze piston rods with piston operating in brass-lined cylinders and brass stuffing boxes. With the plunger pumps the plunger is made, even for hot water, of a hard cast iron. Compound steam ends are seldom used on boiler-feed pumps except in very large plants.

# CHAPTER X.

## FEED-WATER HEATERS AND ECONOMIZERS.

**Value of Feed-water Heaters.** — Exhaust-steam feed-water heaters are used to heat the water fed to boilers with steam exhausted by engines and pumps, and the saving due to their use is so great that plants are seldom constructed without them. Generally speaking, for every 11 degrees that feed water is warmed there is a saving of 1 per cent in the fuel burned. With sufficient exhaust steam available, cold feed water at 70° F. can be raised in temperature to 200 degrees, thus saving nearly 12 per cent of the fuel. The heater will in many locations, therefore, reduce the fuel consumption enough to pay for itself in a few months, the exact time depending upon the cost of the fuel.

**Types of Heaters.** — Exhaust-steam feed-water heaters are of two types, the open and the closed. In the former, the heater consists of a box-like receptacle of cast iron or boiler steel to which the exhaust steam is led so as to fill its interior. The cold feed water is admitted at the top and in most designs trickles to the bottom over a series of trays and is thus brought in contact with the steam and is heated by it. If the water contains scale-forming salts which are precipitated at temperatures below 200 degrees, they collect on the trays, which are removable for cleaning. This type of heater sometimes has a filter in its base for further removing these precipitated salts and impurities that can be intercepted by filtration. The feed water is drawn from a point near the bottom of the heater and pumped to the boilers. The pump must be located so as to receive the water under pressure, as it will not lift the water if it is under a high temperature. The condensation from heating systems, the discharge from traps connected to high-pressure drips, engine jackets, reheaters, etc., which are to be returned to the boilers, can be connected to a heater of this type. A certain water level is maintained in the heater, and to do this the supply of cold feed water is controlled by a valve operated by a float in the heater,

so that cold water is admitted when the boiler-feed pump draws water from it faster than it is supplied by the heating returns, drips, etc. An overflow controlled by a valve operated by a float in the heater is also provided. It is absolutely essential that an efficient grease separator be placed in the steam pipe leading to an open heater, so that oil cannot enter it and pass on to the boilers mixed with the feed water.

Closed heaters usually consist of a cylindrical shell of cast iron or boiler steel containing tubes extending from one head to the other, or coils of pipe. Sometimes the exhaust steam is admitted to the shell so as to surround the pipes or coils containing the feed water, in which event the heater is said to be of the water-tube type. If the steam is inside of the coils with the water surrounding them, it is a steam-tube heater. Closed feed-water heaters are usually supplied with a steam inlet and outlet, although they are sometimes arranged with only a single connection, reliance being placed on the vacuum that forms in the heater when steam is condensed to cause more steam to flow into it. As a certain amount of air exists in the exhaust steam, this will find its way into the heater and is apt to impair its efficiency. For that reason many engineers believe that there should be a double-steam connection, unless some means of removing the air is provided, to prevent air from accumulating and thus to insure a thorough circulation of steam through the heater. Experience has shown in water-tube heaters that the best results are obtained when the water is compelled to pass through the tubes successively, for the reason that the velocity of the water per unit of heating surface is greater. The multipass heater is the outcome of this experience. Closed heaters are made to rest in a horizontal or vertical position. The latter are to be preferred if it is convenient to use them, as they take up much less floor area and the circulation of water in them is much more thorough unless the tubes are subdivided into groups.

**Uses for Heaters.** — The manner in which feed-water heaters are used depends upon the type of plant in which they are installed. With noncondensing engines, where the exhaust steam is available for feed-water heating, the feed water may be raised to a temperature of about 205° F.; and when this is done, it is usually the custom to allow one square foot of heating surface in brass or copper pipes for every 90 pounds of water passed

through the heater per hour. With condensing engines it is sometimes the practice to place a feed-water heater of the closed type in the exhaust pipe of the engine, unless two different sets of conditions exist. If the air pumps and boiler-feed pumps are steam driven, and the steam exhausted by them added to the steam exhausted by all other pumps or steam auxiliaries in the plant amounts to one-seventh or more of the steam generated in the boilers, this exhaust steam from the auxiliaries can be led to an auxiliary feed-water heater and the feed water warmed by it from about 75 to about 205 degrees. In this event, a heater in the exhaust pipe of the engine would be of no value; nor would it, if the feed water were drawn from the condenser discharge, for in that event it would be at nearly as high a temperature as the exhaust steam from the main engine, so that it would hardly pay to attempt to use this steam for heating. The exhaust steam from the main engine being under a partial vacuum, say 26 inches, would have a temperature of about 125 degrees, and the condenser discharge probably about 100 degrees. If the exhaust steam from the auxiliaries be not sufficient in quantity to raise the feed water from a temperature of 60 or 70 degrees to about 205 degrees, then a feed-water heater can be placed in the exhaust pipe of the engine and the water warmed from the lower temperature to about 115 degrees and finally passed through an auxiliary heater and there warmed to as high a temperature as possible with the auxiliary exhaust steam available. This auxiliary heater is almost invariably used in the latest plants while only occasionally is a heater placed in the main exhaust pipe of a condensing engine. Auxiliary heaters of the closed type should have the same amount of heating surface as heaters receiving steam at atmospheric pressure. This rating was previously given. Heaters placed in the exhaust pipes of condensing engines should have more surface, usually one square foot for every 60 pounds of water passed through the heater per hour, this being necessary because the difference between the temperature of the steam and the mean water temperature is less than it is with heaters using steam at atmospheric pressure.

**Condensation in Heaters.** — The condensation that occurs in a feed-water heater amounts to about one-seventh of the weight of the feed water passing through the heater, when the water is raised from 70 to about 205 degrees. In an open heater this, of

course, mingles with the feed water and passes on to the boilers with it. In a closed heater, the steam not being in contact with the water, the condensation occurring has to be removed from the heater, and this can be done by leading it to a steam trap. It should not be forgotten that the steam pressure in a heater is frequently at practically the same pressure as the atmosphere, hence the discharge from the trap should be carried downward, or on a level rather than upward, as there is no pressure to lift the water. It is not necessary to save the condensation in a closed heater unless the cost of this water which, as has been said, is about one-seventh of the total feed water, is sufficient.to make the saving of it advisable; or unless there is not enough exhaust steam to warm the feed water to as high a temperature as may be possible. If there is plenty of exhaust steam, that would otherwise go to waste, available for heating, the heat in this condensation in the heater is of no value. If the condensation from a closed feed-water heater is to be returned to the boilers, it should be trapped, and the discharge from the trap led to a receiving tank combined with a pump controlled by the level of the water in the tank. Other drips from cylinder jackets, high-pressure steam pipes, reheating receivers, etc., can also be led to this tank, each discharging through a trap, or the condensation can be returned to the boilers direct by the steam loop or the Holly system.

**Purchasing Heaters.** — In purchasing a feed-water heater for a power plant for a manufacturing or electric company, the heater is usually purchased by the owner direct, although in some plants, notably those in office buildings, the purchase, delivery, and installation of the feed-water heater is frequently included in the specifications for the piping. The writer believes that it is much better to make the purchase of the heater a separate contract. The specifications for a closed heater should state the amount of heating surface required, the kind of metal that the tubes are to be made of, whether it is to be of the steam or water-tube type, and horizontal or vertical. If the supplying and installation of the heater is to be made part of the contract for the steam piping or some other part of the plant, the contractor undertaking it should be required to deliver and erect it. If the heater is to be purchased direct from its manufacturers, the specifications should ask that the bids cover the delivery of

the heater free on board cars at the nearest railway point to the power plant. The specifications should ask each bidder to furnish a blue-print or drawing showing the details of the construction of the heater it is proposed to furnish, its exact dimensions and the location of the steam and water outlets and their sizes.

There is no general rule that the writer is aware of for proportioning heaters of the open type. Therefore a specification for a heater of this kind should state the quantity of water to be passed through the heater in a given time and also the initial temperature of the water and the final temperature desired. A print or drawing of the heater a bidder is to provide should be obtained and the inside dimensions of the heater, the volume of the steam and water space, etc., should be investigated. The heater must not be too small, otherwise the water in passing through it cannot be broken up in sufficiently small particles to allow the water to mix thoroughly with the steam and thus be sufficiently heated by it. If a guarantee as to the efficiency of a feed-water heater is required, it should be in the following form: The maker guarantees the heater to be capable of warming —— pounds of water per hour from a temperature of —— degrees F. to —— degrees F. with sufficient steam at atmospheric pressure.

**Economizers.** — Economizers are used in connection with steam boilers to warm water either for boiler feeding or for some manufacturing purpose, by the heat in the gases from boilers that would otherwise go to waste. Economizers are also useful in increasing the capacity of a boiler plant already in operation, in providing means for storing a large quantity of water at high temperatures, which is of advantage in the event of a sudden increase in the demand for steam. They also deliver the water to boilers at high temperature and reduce strains in boilers due to the admission of cold water. One disadvantage of economizers is that they reduce the draft slightly owing to the friction of the gases passing through them and to the reduction in temperature of the gases. Provision should be made for this by the use of mechanical draft or of larger chimneys than would be necessary if economizers are not used. In plants already constructed the addition of economizers reduces the coal consumption, and this counterbalances in a measure the loss in draft, as less draft is necessary with them because less fuel is burned.

Economizers consist of vertical cast-iron pipes about 4 inches in diameter and 9 feet long placed in rows several inches apart, each row being connected at the top and bottom to cast-iron headers through which the water is supplied and withdrawn. They are provided with scrapers that encircle the pipes and that are

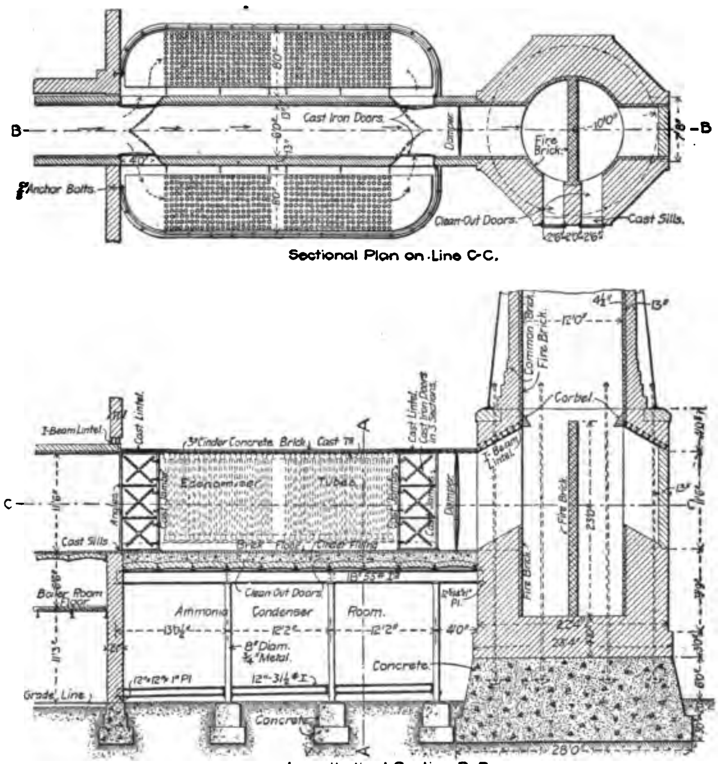



Fig. 51. Economizer Arrangement, Plant of Schwarzchild and Sulzberger, Chicago, Ill. (L. Levy, Chief Engineer.)

continuously raised and lowered by a suitable mechanism, the power coming usually from a small steam engine. Economizers are placed in brick flues between the boilers and the chimney, and a by-pass flue must be provided for use when the economizer is out of service, either for cleaning, which is necessary at intervals, or for repairs. Economizers are used to a very considerable extent in Europe, more so in fact than in the United States, where

engineers are now, however, rapidly appreciating their advantages. When first introduced, a number of failures occurred mainly due to improper design. They were then either constructed of poor materials improperly put together or there was a lack of sufficient heating surface necessary to make the saving that they should produce. Since the economizer has been perfected and made of durable materials, their advantages have become recognized.

The heating surface in economizers takes the place, in a measure, of additional boiler-heating surface. If a boiler is operating under certain conditions so that, with a certain rate of evaporation per square foot of heating surface, a certain temperature of waste gases follows, it would be possible to add more heating surface and abstract more heat from these gases; but the boiler-heating surface would not be so efficient in doing this as an equivalent amount of economizer surface, for the reason that the average temperature of the water in the economizer is lower than that of the water in the boiler; this causes a more rapid transfer of heat from the gases to the water in the economizers than there would be from the gases to the water in the boiler. The saving that an economizer can produce increases as the flue temperature increases because there is more heat for it to absorb. An economizer can only heat the water to a given point, hence as the temperature of the entering water increases, due to previous heating, as in an exhaust steam heater, the saving possible with the economizer becomes less. If the ratio of boiler-heating surface to the amount of coal burned is such that as much of the heat in the waste gases is absorbed as is possible, there is not as much heat left in the gases for the economizers to save as there would be if the ratio of boiler-heating surface to the amount of coal burned were less. Barrus, in his book on "Boiler Tests," states in effect that where a boiler is operated most efficiently, the temperature of the waste gases should not exceed about 400° F., and boilers are usually so proportioned that about this temperature exists when the highest efficiency is attained. Owing, however, to the accumulation of soot upon the boiler surface or to running at a higher rate of evaporation than the normal, the temperature of the waste gases usually exceeds that given, so that an economizer is valuable in even a well-proportioned boiler plant. Mr. Barrus gives some excellent data showing the saving made by economiz-

ers with low temperatures of flue gases, and they are reproduced in Table 25, herewith.

TABLE 25. — BARRUS' TESTS OF ECONOMIZERS.

| | | | | |
|---|---|---|---|---|
| Heating surface, boiler, sq. ft. | 1894 | 1058 | 5592 | 3126 |
| Heating surface, economizer, sq. ft. | 1600 | 1920 | 1280 | 1600 |
| Temperature of gases leaving boiler, deg. | 376 | 361 | 403 | 435 |
| Temperature of gases leaving economizer, deg. | 231 | 254 | 299 | 279 |
| Temperature of feed water entering economizer, deg. | 95 | 79 | 111 | 84 |
| Temperature of feed water entering boiler, deg. | 175 | 145 | 169 | 196 |
| Increased evaporation produced by economizer, per cent | 10.5 | 7 | 9.3 | 12.8 |

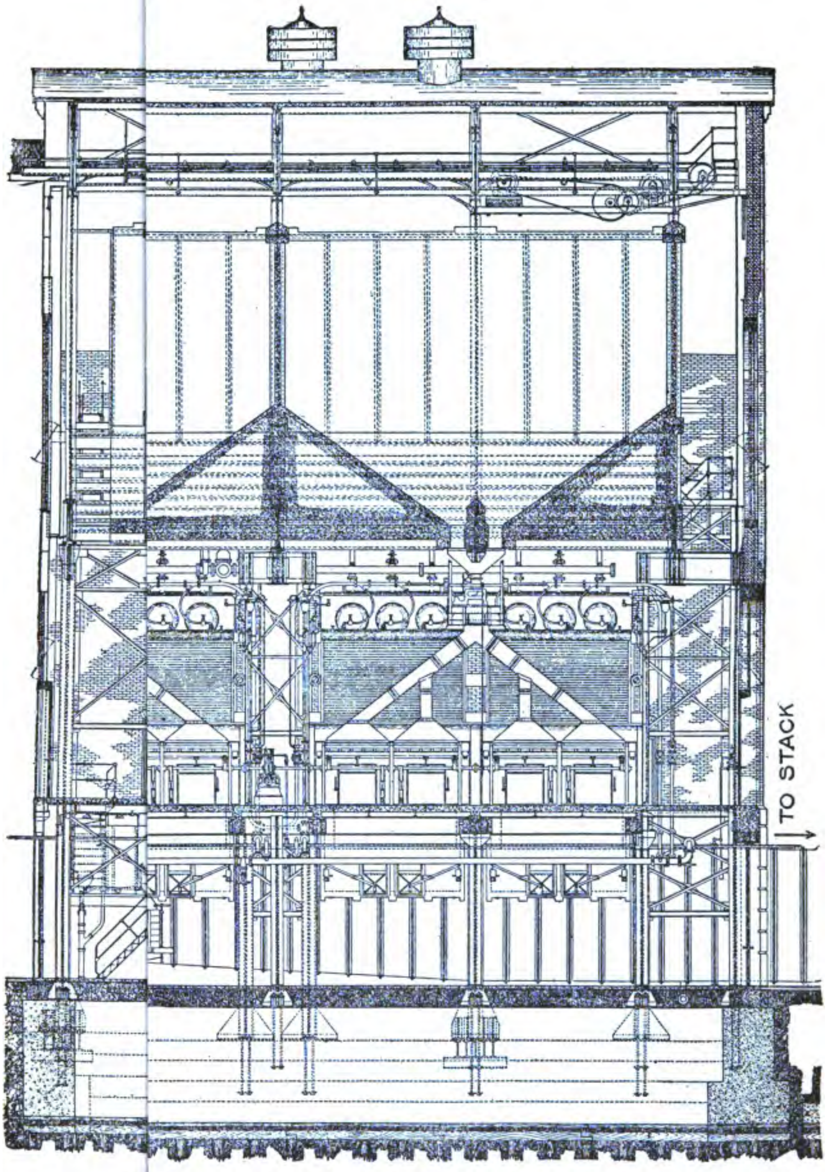
Mr. William R. Roney, in a paper on "Mechanical Draft" (Transactions, Am. Soc. M. E., Vol. XV), gives some additional data on the saving due to economizers working under various conditions in nine different plants. They are given in Table 26.

TABLE 26. — RONEY'S TESTS OF ECONOMIZERS.

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Plants tested | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 |
| Gases entering economizer, deg. | 610 | 505 | 550 | 522 | 505 | 465 | 490 | 495 | 595 |
| Gases leaving economizer, deg. | 340 | 212 | 205 | 320 | 320 | 250 | 290 | 190 | 299 |
| Water entering economizer, deg. | 110 | 84 | 185 | 155 | 190 | 180 | 165 | 155 | 130 |
| Water leaving economizer, deg. | 287 | 276 | 305 | 300 | 300 | 295 | 280 | 320 | 311 |
| Gain in temperature of water, deg. | 117 | 192 | 120 | 145 | 110 | 115 | 115 | 165 | 181 |
| Fuel saving, per cent | 16.7 | 17.1 | 11.7 | 13.8 | 10.7 | 11.2 | 11.0 | 15.5 | 16.8 |

In considering the saving that an economizer produces it is necessary to take into account the interest on the investment and the cost of repairs, cleaning, etc. Economizers cost, it is said, about $5.40 per boiler horse-power for plants of 1000 boiler horse-power or over, on the basis of 4.8 square feet of economizer surface per boiler horse-power. This includes the cost of the brick setting, delivering, and erecting, etc. Three per cent of the investment will probably do more than pay for the cost of the operation, cleaning, and repairs. Assume a 1000-horse-power boiler plant to operate 300 ten-hour days per year, that the coal consumption is $3\frac{1}{2}$ pounds per boiler horse-power per hour and that coal costs $3 per ton of 2000 pounds delivered. The annual fuel cost would then be $15,750, and if an economizer reduced this by 12 per cent, then the saving that an economizer would

produce would be $1890. The cost of the economizer at $5.40 per boiler horse-power would be $5400 and 8 per cent of this for interest, repairs, operating, and cleaning is $432. Deducting $432 from $1890 would leave a net saving of $1458, which is sufficient to pay for the economizer in less than four years. If the plant was operated continuously, the annual fuel cost would be $45,990 and the net saving $5085, sufficient to pay for the economizer in about one year. The 12 per cent that was assumed in the previous calculations is a conservative estimate for the saving with a low temperature of feed water. In the four tests made by Mr. Barrus, previously noted, the average saving was 9.9 per cent and this was obtained with unusually low temperatures of the escaping gases, due partly to the low-steam pressure under which the plants were operated, these varying in the different tests from 68 to 82 pounds. Economizer makers will guarantee that they can produce a saving of 6½ per cent when the temperature of the water entering them is as high as 200 degrees, the economizers having 4.5 square feet of heating surface per boiler horse-power and the boilers working at their normal rating.

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## CHAPTER XI.

### MECHANICAL DRAFT.

**Theory of Combustion.** — As pointed out in a previous chapter, there are two independent factors that effect the efficiency of a steam boiler. One is the efficiency of the heating surface or its ability to transmit to the water the heat to which it is exposed, and the other is the efficiency of the furnace, by which is meant the amount of heat actually contained in each unit volume of furnace gas compared to that theoretically possible of attainment. A high surface efficiency demands that the temperature of the gases in contact with the boiler be as high as possible, for the rate of heat transfer is some function of the difference in temperature of the water and gases, and the greater the difference the more heat per unit of surface will be transmitted. A high furnace efficiency demands that just the proper amount of air be supplied to the furnace per pound of fuel burned. Too little air, due to a too thick bed of fuel on the grate in proportion to the available draft, results in loss due to the incomplete combustion of the fuel, part of the carbon being burned only to carbonic oxide instead of to carbonic acid, as it should if the combustion is complete. Too much air, on the other hand, due to too thin a bed of fuel, lowers the temperature of the furnace gases and thereby decreases the heat transferred per unit of heating surface, owing to the smaller difference between the gas and the water temperatures; it also increases the volume of gases and this in turn necessarily increases the velocity of their flow through the passages of the boiler, thus giving less time for their heat to be absorbed.

A thin fire allows an excess of air to enter the furnace, while a thick fire tends to reduce it on account of the greater resistance offered. Mr. Walter B. Snow writes as follows in explaining the higher efficiency of a high rate of combustion: "If the size of a grate is reduced but the same amount of fuel burned by increasing the rate of combustion, the diminished area of the

grate and of the exposed interstices between the fuel necessitates a higher velocity to secure the admission of a given volume of air. This increased velocity in turn requires greater draft or air pressure. The volume at any given temperature passing through the coal is proportional to the velocity, but the pressure varies as the square of the velocity. Therefore, if a given grate be reduced one-half and the rate of combustion doubled, so as to maintain the same total consumption, the same volume of air would have to travel through the exposed interstices at twice the velocity. But the pressure or vacuum would be four times as great, and, as a consequence, the air would be forced or drawn into spaces between the fuel which it could not reach under lesser impelling force. Much more intimate contact and distribution are the results. Less free oxygen passes through the fuel bed unconsumed, and for a given supply of air a higher efficiency of the fuel is attained." Most leading authorities unite in the belief that a higher efficiency is secured in steam boilers when operating at comparatively high rates of combustion. It is advisable to provide for this in plants using coal that can be burned rapidly, care being taken at the same time to provide sufficient draft to run the boilers over their normal rating when occasion demands. Certain coals high in ash and in sulphur cannot be burned at more than ordinary rates for reasons explained in Chapter II in the section relating to coals.

**Necessity for Ample Draft.** — Means for providing a strong draft for steam boilers is one of the most essential features of a plant, for if sufficient coal can be burned a boiler will generate steam several times as rapidly as under normal conditions. This means that a smaller number of the boilers can be relied upon to furnish the steam required, while others are shut down for repairs or cleaning, than would be needed if less draft was available. The investment for boilers, therefore, need not be so great. Of course, when making steam at abnormally high rates of evaporation, the efficiency is not so good as under the conditions of normal working, but it is cheaper to allow some fuel to go to waste occasionally than to spend more money in the first place for boilers only used at long intervals. More complaints with steam boilers are probably traceable to insufficient draft than to any other cause. Draft is secured by a chimney, by mechanical draft produced by fans, or by steam-jet blowers.

**Mechanical Draft.** — Mechanical draft may be secured by two methods, known as induced draft and forced draft. In the former, a fan is connected with the smoke flue from a boiler or batteries of boilers, so as to suck the gases through the furnace, gas passages, and flues to the fan, which discharges it usually through a short chimney, but sometimes through a high one. Forced draft is the term applied to that system where air is forced into the furnace beneath the grate bars, either by a fan or by a steam-jet blower. The objection to forced draft lies in the fact that a pressure greater than that of the atmosphere is created in the furnace and gas passages which may cause the gas to pass outward through cracks in the setting of boilers and through the fire doors when the fires are being replenished or cleaned. Dampers in the blast pipe to the ashpit, to be closed when the fire doors are opened, may overcome this latter objection. With induced draft the leakage through the doors and brickwork is of course inward. Both induced draft and forced draft are used to a considerable extent, the choice usually depending upon which is the cheapest and easiest to install.

Steam-jet blowers in which a jet of steam is relied upon to induce a flow of air into the ashpit of a boiler can only produce a moderate draft. Their steam is believed to be useful in preventing clinker from forming on the grates with certain kinds of anthracite coals. Steam jets are used to a considerable, but decreasing, extent in the anthracite coal regions. They can oftentimes be applied to existing plants with advantage, where mechanical draft would be difficult to install. Experiments made by a Board of Steam Engineers of the Navy Department with five different types of steam jets showed that they used from 8.3 to 21.2 per cent of the steam generated in the boiler to which the jets were applied. Draft can be obtained from fans with only a fractional part of these amounts. Under ordinary conditions a fan for mechanical draft can be driven by from 1 to 5 per cent of the steam evaporated in the boilers operating under ordinary conditions, depending upon the size of the plant.

**Advantages of Mechanical Draft.** — Draft produced by fans possesses many advantages over chimneys as ordinarily proportioned. Probably the greatest of these is its flexibility, it being possible to regulate the speed of the fan so that the proper rate of combustion for the amount of steam required is maintained

entirely independent of the weather conditions; another important advantage is the ability of the fans to create a much greater draft than is possible with a chimney. Steam engines for driving fans are frequently fitted with valves arranged to govern the speed of the engine according as the boiler pressure

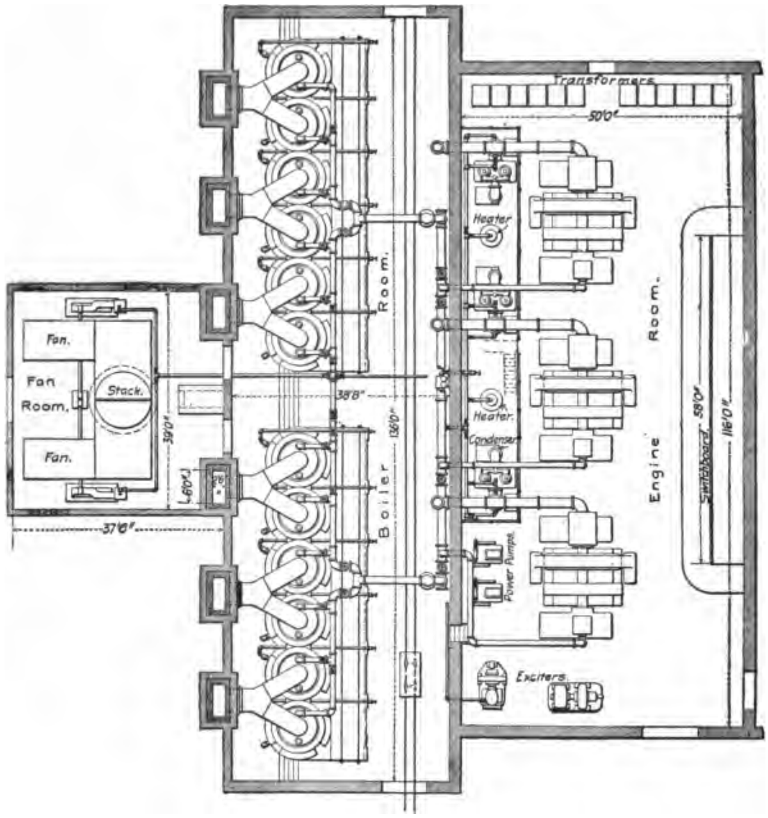



Fig. 52. Power House, Olympia Mills, Columbia, S. C.
(M. B. Smith-Whaley, Engineer.)

varies, increasing it as the pressure falls and reducing it as it rises above the normal. Mechanical draft enables economizers to be placed in the flue and reduce the temperature of the escaping gases, by heating the feed water, far below the temperature that is necessary in a chimney to create a draft. The reduction in draft due to the use of economizers is a much greater per-

centage of the available draft with a chimney than it is of the draft where fans are employed. Again, the greater draft of fans enables cheap low-grade fuels to be burned that could not easily be used with the chimney draft, and the saving that these fuels brings about in some localities is a very considerable sum of money. Still another point in favor of mechanical draft lies in the portability of the fans in case a change of location is desired.

**Theory of Fans.** — Before explaining the method of selecting a fan for a given service it is necessary to consider briefly their theory of operation. Fans for mechanical draft are invariably of the centrifugal type and consist of a paddle wheel revolving in a sheet-iron casing, the air entering an opening in the casing at the axis of the wheel and being propelled radially to the casing by the action of centrifugal force, finally escaping by an outlet provided. The velocity with which the air is moved is expressed by the well-known formula:

$$v = \sqrt{2\,gh}, \tag{1}$$

in which $v$ equals the velocity and $h$ equals the head due to the velocity and also to the pressure divided by the density. Therefore:

$$v = \sqrt{(2\,gp \div d)}. \tag{2}$$

The work done in moving this air is equal to the product of the velocity of the air in feet per second, the pressure in pounds per square foot, and the effective area in square feet over which this pressure is exerted. If $W$ represents the work done, $p$ the pressure in pounds per square foot, $a$ the area in square feet, and $v$ the velocity in feet per second, then:

$$W = pav. \tag{3}$$

From (2) we have, by squaring and transposing:

$$p = dv^2 \div 2\,g; \tag{4}$$

and substituting its value in (3) we have:

$$W = dav^3 \div 2\,g. \tag{5}$$

From this it will be seen that the power required to drive a fan varies as the cube of the velocity. In other words, if the velocity

is doubled the power required will be eight-fold; if tripled, twenty-seven fold. As the velocity of the air is practically the same as the peripheral speed of the fan it will be seen how essential it is to use a large fan at its proper speed rather than a small fan running at a higher speed than is necessary to obtain the desired pressure, and thus the desired volume. The pressure varies as the square of the speed, as shown in formula (4), hence the pressure is quadrupled by doubling the speed.

**Design of Fans.** — From the work upon "Mechanical Draft," by Mr. Walter B. Snow, published by the B. F. Sturtevant Company, the following quotations relating to the design of fans have been taken:

"In the design of a wheel to meet given requirements, it is necessary to make its peripheral speed such as to create the de-

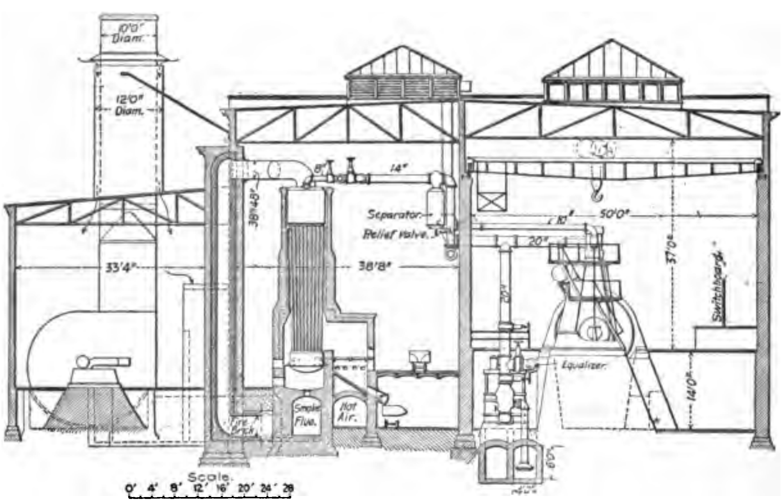



Fig. 53. Cross Section, Power House, Olympia Mills.

sired pressure, and then so proportion its width as to provide for the required air volume. Evidently the velocity and corresponding pressure may be obtained either with a small wheel running at high speed or a large wheel running at low speed. But if the diameter of the wheel be taken too small, it may be impossible to adopt a width, within reasonable limits, which will permit of the passage of the necessary amount of air under the desired pressure. Under this condition it will be necessary to run the

fan at higher speed in order to obtain the desired volume. But this results in raising the pressure above that desired, and in unnecessarily increasing the power required. On the other hand, if the wheel be made of excessive diameter it will become almost impracticable on account of its narrowness. Between these two extremes a diameter must be intelligently adopted which will give the best proportions.

" It has been determined experimentally that a peripheral discharge fan, if enclosed in a case, has the ability, if driven to a certain speed, to maintain the pressure corresponding to its tip velocity over an effective area which is usually denominated the 'square inches of blast.' This area is the limit of its capacity to maintain the given pressure. If it be increased the pressure will be reduced, but if decreased the pressure will remain the same. As fan housings are usually constructed, this area is considerably less than that of either the regular inlet or outlet.

" The square inches of blast, or, as it may be termed, the capacity area of a cased fan, may be approximately expressed by the empirical formula:

$$\text{Capacity area} = DW \div X.$$

In which $D$ = diameter of fan wheel in inches..
$W$ = width of fan wheel at circumference in inches.
$X$ = a constant dependent upon the type of fan and casing.

" An approximate value for $X$ for Sturtevant fans for general practice is not far from 3, but this is to be used only to determine the capacity area over which the given pressure may be maintained. This is not a measure of the area of the casing outlet, which is always larger than the square inches of blast. As a consequence, the pressure is lower and the volume discharged is somewhat greater than would result through an outlet having the square inches of blast for its area. But the maximum pressure may be realized when the sum of resistances is equivalent to a reduction of effective outlet area to that of square inches of blast. The volume of air which, under the given pressure, will flow through the given capacity area, and hence the volumetric capacity of the fan under the given conditions, may be determined from Table 27. In a similar manner the horse-power may be ascertained, the proper efficiency coefficient being applied.

"Both the volume and the power required will evidently increase with the area of the outlet, being greater with the normal outlet than with that representing the capacity area. But this increase will not be proportional to the area, for the pressure and consequently the velocity will be lower with the larger area. The greatest delivery of air and the largest consumption of power will occur when the casing is entirely removed and the fan left free to discharge entirely around its periphery.

"If volume alone regardless of pressure is the requisite, the larger the fan the less the power required. There is a strong temptation, however, for a purchaser to buy a smaller fan and run it at a higher speed; for he sees only the first cost and does not realize the entailed expenditure for extra power. If possible a fan should never be made so small that it is necessary to run it above the required pressure in order to deliver the necessary volume. To double the volume under such circumstances requires eight times the power; three times the volume demands twenty-seven times the power.

"When a fan is employed for exhausting hot air or gases, the speed required to maintain a given pressure difference is evidently greater than that necessary when cold air is handled, the difference being due to, and inversely proportional to, the absolute temperature."

With forced draft it will be safe to assume that 300 cubic feet of air are supplied per pound of coal burned, and with induced draft 300 cubic feet supplied but 600 cubic feet exhausted by the fan if the gases are to be at a temperature of about 500° F. or 450 cubic feet exhausted if at 300 degrees temperature, as might be expected with economizers. With induced draft the fan has to handle hot gases of approximately double the volume that it would if the fan was supplying air to the grates. As the gases are lighter, however, the power required per cubic foot of gas removed is less than it is per cubic foot of air with forced draft. Knowing the amount of coal burned under ordinary conditions, say four pounds per boiler horse-power per hour, the amount of air required per minute can be determined. The draft that should be available should be that required to overcome the resistance to the passage of air through the grates and of the gases through the boiler smoke flue and chimney and through an economizer if one is used. The pressures usually required in

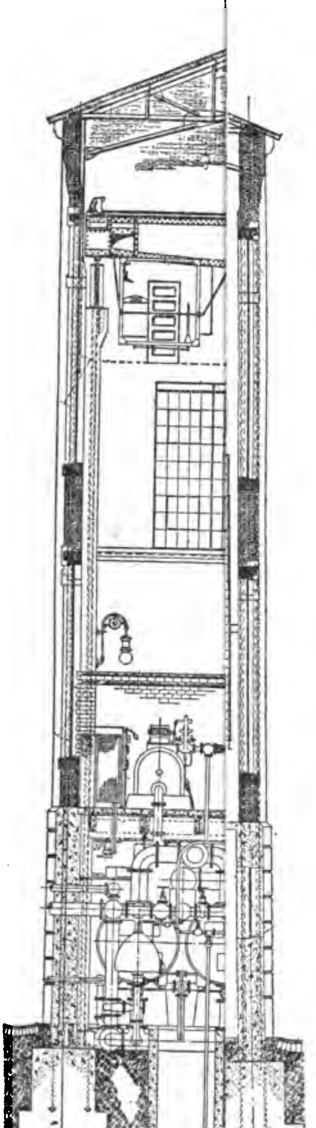


PLATE 18.—CROSS S ROOMS,

forced draft vary from $\frac{1}{2}$ to 1 ounce per square inch and in induced draft from $\frac{1}{2}$ to $\frac{3}{4}$ ounce, depending on the fineness of fuel, the readiness with which it burns, the length of flues and number of bends in them, etc. A pressure of one ounce per square inch is equal to 1.73 inches of water at temperatures of 50° F.

Tables 27 and 28 are taken from "Mechanical Draft," before mentioned. In the former is given the volume of air in cubic feet for various pressures which may be discharged in one minute through an orifice having an effective area of discharge of one square inch, or per square inch of blast. The method of using these tables can best be explained, perhaps, by working out a typical case. Suppose we are to design an induced-draft fan to handle the gases from 1000 horse-power of boilers against a pressure of $\frac{3}{4}$ ounce. At 4 pounds of coal per horse-power per hour and 600 cubic feet of gas per pound, 40,000 cubic feet of gas would have to be exhausted per minute. Opposite $\frac{3}{4}$-ounce pressure in Table 27 it will be seen that a fan will supply 31.06 cubic feet, say 31 cubic feet, per square inch of blast. Dividing 40,000 by 31 would give 1290 as the number of square inches of blast the fan would require. But the square inches of blast equal $DW \div 3$. In standard fans of the Sturtevant make $W$ is approximately equal to $D \div 2.4$, therefore:

$$\text{Square inches of blast} = D^2 \div 7.2.$$

TABLE 27. — VOLUME OF AIR DISCHARGED AND HORSE-POWER REQUIRED WHEN AIR UNDER GIVEN PRESSURE IS ALLOWED TO ESCAPE INTO THE ATMOSPHERE.

| Pressure in ounces per square inch. | Volume of air discharged through an orifice of an effective area of discharge of 1 square inch (or per square inch of blast), cubic feet per minute. | Horse-power required to move the given volumes of air under given conditions of discharge. |
|---|---|---|
| $\frac{1}{4}$ | 17.95 | 0.00122 |
| $\frac{3}{8}$ | 21.98 | 0.00225 |
| $\frac{1}{2}$ | 25.37 | 0.00346 |
| $\frac{5}{8}$ | 28.36 | 0.00483 |
| $\frac{3}{4}$ | 31.06 | 0.00635 |
| $\frac{7}{8}$ | 33.54 | 0.00800 |
| 1 | 35.85 | 0.00978 |
| $1\frac{1}{8}$ | 38.01 | 0.01166 |
| $1\frac{1}{4}$ | 40.06 | 0.01366 |
| $1\frac{3}{8}$ | 42.0 | 0.01577 |
| $1\frac{1}{2}$ | 43.86 | 0.01794 |

Substituting 1290 for the square inches of blast we have: 1290 $= D^2 \div 7.2$, from which $D$ equals 96 inches or 8 feet for the diameter of the fan.

To determine the velocity at which the fan should be run, reference should be made to Table 28. For the column corresponding to $\frac{3}{4}$ ounce it will be seen that an 8-foot fan should run at 178 revolutions per minute to give the capacity required. By increasing the speed at times of overload the pressure and volume of gases can be increased.

In the last column of Table 27 is given the horse-power required per square inch of blast to move given quantities of air, under different pressures and at a temperature of 50° F. For forced draft these quantities can be used directly, but it should be remembered that they refer to the theoretical power required to move the air and they should be at least doubled to allow for

TABLE 28. — REVOLUTIONS OF FAN OF GIVEN DIAMETER NECESSARY TO MAINTAIN A GIVEN PRESSURE OVER AN AREA WHICH IS WITHIN THE CAPACITY OF THE FAN.

| Diameter of fan wheel, in feet. | Pressure, in ounces per square inch. | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | $\frac{1}{8}$ | $\frac{1}{4}$ | $\frac{3}{8}$ | $\frac{1}{2}$ | $\frac{5}{8}$ | $\frac{3}{4}$ | $\frac{7}{8}$ | 1 | $1\frac{1}{8}$ | $1\frac{1}{4}$ |
| 3 | 194 | 274 | 336 | 388 | 433 | 475 | 513 | 548 | 581 | 612 |
| $3\frac{1}{2}$ | 166 | 235 | 288 | 332 | 372 | 407 | 439 | 469 | 498 | 525 |
| 4 | 146 | 206 | 252 | 291 | 325 | 356 | 384 | 411 | 436 | 459 |
| $4\frac{1}{2}$ | 129 | 183 | 224 | 258 | 289 | 316 | 342 | 365 | 387 | 408 |
| 5 | 116 | 164 | 202 | 232 | 260 | 285 | 308 | 329 | 349 | 367 |
| $5\frac{1}{2}$ | 106 | 149 | 183 | 211 | 236 | 259 | 280 | 299 | 317 | 334 |
| 6 | 97 | 137 | 168 | 194 | 217 | 238 | 256 | 274 | 290 | 306 |
| $6\frac{1}{2}$ | 90 | 126 | 155 | 179 | 200 | 219 | 236 | 253 | 268 | 282 |
| 7 | 83 | 117 | 144 | 166 | 186 | 203 | 220 | 235 | 240 | 262 |
| $7\frac{1}{2}$ | 78 | 110 | 135 | 155 | 173 | 190 | 204 | 219 | 232 | 245 |
| 8 | 73 | 103 | 126 | 146 | 163 | 178 | 192 | 205 | 218 | 230 |
| $8\frac{1}{2}$ | 69 | 97 | 119 | 137 | 153 | 167 | 181 | 194 | 205 | 216 |
| 9 | 65 | 92 | 112 | 129 | 144 | 158 | 171 | 183 | 194 | 204 |
| $9\frac{1}{2}$ | 61 | 87 | 106 | 123 | 137 | 149 | 162 | 173 | 183 | 193 |
| 10 | 58 | 82 | 101 | 116 | 130 | 142 | 154 | 164 | 174 | 184 |
| 11 | 53 | 75 | 92 | 106 | 118 | 129 | 140 | 150 | 158 | 167 |
| 12 | 49 | 69 | 84 | 97 | 108 | 119 | 128 | 137 | 145 | 153 |
| 13 | 45 | 63 | 78 | 90 | 100 | 110 | 116 | 126 | 130 | 141 |
| 14 | 42 | 59 | 72 | 83 | 93 | 102 | 110 | 117 | 124 | 131 |
| 15 | 39 | 55 | 67 | 78 | 87 | 95 | 102 | 110 | 116 | 122 |

the fan and engine friction and the power required to run the fan, considering it was moving no air and to overcome the friction of the air in the fan casing.

With induced draft the air is of less density on account of its higher temperature, so that less power is required per cubic foot of gas moved. If the gases have a temperature of 600 degrees the quantities given in the last column of Table 27 can be multiplied by 0.5, and by 0.58 if the temperature is 450 degrees, to obtain the actual power required to move the air. The engine driving the fan should have its cylinders so proportioned that it can easily develop power 50 per cent in excess of that calculated.

Table 29 shows the capacity of the American Blower Company's fans used for induced draft, according to its catalogue.

TABLE 29. — INDUCED DRAFT CAPACITY TABLE FOR FANS.

| Size of fan. | Diameter of wheel, inches. | Width at periphery. | Diameter of inlet. | Size of outlet, square inches. | Speed R.P.M., for 1-inch draft. | Capacity of fan in cubic feet per minute, temperature gases 550° F. | H.P. boiler capacity from fan capacity. | Pounds coal per hour at 5 pounds per H.P. hour. | Brake H.P. to drive fan at speed. | Capacity of fan per inch width at periphery. | H.P. per inch width at periphery. | Cubic feet air per minute for combustion. Temperature 62° F. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 50 | 30 | 12½ | 20 | 18 | 740 | 5,030 | 116 | 580 | 2.02 | 403 | .162 | 2,260 |
| 60 | 36 | 14¼ | 23 | 21½ | 615 | 6,900 | 159 | 795 | 2.76 | 485 | .194 | 3,100 |
| 70 | 42 | 16½ | 26 | 24½ | 530 | 9,325 | 215 | 1,075 | 3.72 | 563 | .226 | 4,200 |
| 80 | 48 | 17½ | 30 | 27 | 460 | 11,300 | 261 | 1,305 | 4.50 | 645 | .258 | 5,100 |
| 90 | 54 | 20¾ | 34 | 30½ | 410 | 15,100 | 349 | 1,745 | 6.03 | 728 | .290 | 6,800 |
| 100 | 60 | 23¼ | 38 | 34½ | 370 | 18,750 | 433 | 2,165 | 7.50 | 844 | .322 | 8,450 |
| 110 | 66 | 26 | 42 | 37½ | 335 | 23,000 | 532 | 2,660 | 9.20 | 885 | .354 | 10,370 |
| 120 | 72 | 30¼ | 46 | 41½ | 310 | 29,250 | 677 | 3,385 | 11.72 | 970 | .387 | 13,200 |
| 140 | 84 | 34⅜ | 53 | 48 | 265 | 38,800 | 896 | 4,480 | 15.50 | 1132 | .452 | 17,450 |
| 160 | 96 | 38 | 60 | 54 | 230 | 49,000 | 1130 | 5,650 | 19.60 | 1290 | .516 | 22,000 |
| 180 | 108 | 41¼ | 68 | 60 | 205 | 59,900 | 1385 | 6,925 | 24.00 | 1453 | .581 | 27,000 |
| 200 | 120 | 47 | 76 | 66 | 185 | 75,500 | 1746 | 8,730 | 30.25 | 1610 | .644 | 34,000 |
| 220 | 132 | 50 | 84 | 72 | 170 | 88,600 | 2050 | 10,250 | 35.50 | 1775 | .710 | 40,000 |
| 240 | 144 | 54 | 92 | 78 | 155 | 104,600 | 2420 | 12,100 | 41.80 | 1940 | .775 | 47,000 |

It is based on a temperature of 550° F. for the gases, and air temperature of 62 degrees, on 18 pounds or 234 cubic feet of air per pound of coal, and a consumption of 5 pounds of coal per boiler horse-power. Mr. F. R. Still, of the American Blower Company, recently wrote as follows explaining it: "This table gives the diameter of the fan wheel, width at periphery, diameter of the inlet, size of the outlet, and the maximum speed necessary

to produce a draft of one inch of water, which is about the strongest draft that is ordinarily required. The table also gives the capacity of the fan per inch of width (of blade) at periphery, and the horse-power per inch of width. If it is desired to run the fan at a slower speed than is given opposite any of our sizes, determine on the speed at which the fan is to run, find the capacity of same per inch width and divide the required capacity for a given horse-power in the plant by this capacity per inch, and it will determine how wide the fan should be at the periphery, with the increased diameter and the slower speed. The horse-power can be determined by multiplying this width by the horse-power per inches in width."

## CHAPTER XII.

### CHIMNEYS.

**Size of Chimneys.** — There are two factors that affect the capacity of a chimney, its cross-sectional area and its height. The capacity is proportional to the square root of the height of the chimney, and, for a given height, the capacity is directly proportional to its cross-sectional area within certain limits. Earlier in this book attention has been called to the importance of having a strong draft in order that cheap, low-grade fuels may be burned successfully, and, also, that there may be sufficient reserve capacity in the boilers at times of abnormal demands. In this latter respect ample draft is just as good as additional boiling-heating surface, and, as a general proposition, it costs less. Although an engineer once said that high chimneys "are monuments to the folly of their builders," yet this opinion should not deter one from building chimneys from 150 to 200 feet in height, and for very large plants still higher. In certain localities where smoke or obnoxious gases are objectionable, or where ample draft is required, tall chimneys are necessary. Chimneys of the heights mentioned are desirable where high rates of combustion are to be employed, and the height of the chimney is governed somewhat by the amount of coal that is to be burned per square foot of grate in a given time.

**Height of Chimneys.** — Mr. J. J. De Kinder, an engineer of considerable experience in steam-plant operation, recommends the following heights for chimneys with the coals mentioned: 75 feet for free-burning bituminous coal, 100 feet for slow-burning bituminous slack, 115 feet for slow-burning bituminous coal, 125 feet for anthracite pea coal, 150 feet for anthracite buckwheat coal. These recommendations are probably intended for boiler plants of moderate size only. As a matter of fact the height is governed by the draft required to burn the kind of coal that is to be used as shown in the following section.

**Draft.** — In Chapter II it was shown how a boiler plant should be provided with a grate large enough so that with a given draft sufficient coal could be burned to generate the steam required. Now the amount of draft theoretically available depends upon the height of the chimney, if a chimney is used, and the draft available for promoting combustion is the theoretical draft less the draft used up by the friction of the gases in passing through the boiler, the smoke flue, and the chimney. It is proposed in this chapter to give some idea of the friction due to the passage of air through fuels of different kinds at different rates of combustion, and the friction in boilers, smoke flues, and chimneys, so that it is possible to obtain a proper relation between grate area, flue area, and stack dimension.

Table 30 gives the theoretical draft in inches of water that will be obtained from a chimney 100 feet high for various outside temperatures and for various temperatures of waste gases, the table being based upon a barometric pressure of 30 inches.

TABLE 30. — THEORETICAL DRAFTS IN CHIMNEYS.

| Chimney temperature. | Temperature external air. | | |
|---|---|---|---|
| | 30° | 60° | 90° |
| 300° | .511 | .420 | .340 |
| 400° | .632 | .541 | .461 |
| 500° | .730 | .639 | .559 |

Gas temperatures of from 400 to 500 degrees can be assumed for most power boilers if run at their usual rating, and the outside temperatures can be taken at 90 degrees if the full load obtains in the summer time. The theoretical draft varies directly as the height of chimney, hence the draft for other heights can be obtained by direct proportion.

The draft required to move the air through the fires varies with the kind of coal, and with the thickness of the bed of fuel on the grate. The Stirling boiler catalogue gives data, from which Table 31 has been prepared, showing the draft in inches of water required to burn different kinds of coal at the various rates of combustion under average conditions.

TABLE 31.—DRAFT IN INCHES OF WATER TO BURN DIFFERENT COALS.

| Pounds coal per square foot grate per hour. | 10 | 15 | 20 | 30 | 40 |
|---|---|---|---|---|---|
| No. 3, buckwheat anthracite | .40 | .74 | 1.25 | ...... | ...... |
| No. 1, buckwheat anthracite | .23 | .43 | .68 | ...... | ...... |
| Anthracite pea | .16 | .30 | .45 | .88 | ...... |
| Run of mine semibituminous | .08 | .13 | .20 | .37 | .62 |
| Bituminous slack | .07 | .11 | .16 | .27 | .42 |
| Run of mine bituminous | .05 | .08 | .10 | .17 | .27 |

The friction of the gas passages in a Babcock and Wilcox boiler and a Stirling boiler when run at rating is about 0.2 inch of water, and at 50 per cent overload about 0.40 inch. The friction of a horizontal tubular boiler is about the same.

A common method of proportioning flues is to allow 35 square feet of sectional area per thousand horse-power in boilers. This is hardly a correct method as the frictional loss does not vary with the cross section. Table 32 has been prepared giving the sectional area of flue required for different boiler capacities for

TABLE 32.—FRICTIONAL LOSS IN SQUARE IRON FLUES IN INCHES OF WATER PER 100 FEET OF LENGTH.

| Horse-power. | 100 | 200 | 300 | 400 | 500 | 600 | 800 | 1000 | 1500 | 2000 | 2500 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Area flue, sq. ft. for .05 in. frictional loss | 3.9 | 6.6 | 9.0 | 11.2 | 13.5 | 15.4 | 19.5 | 22.8 | 31.0 | 38 | 46 |
| Friction sharp bend | .18 | .245 | .305 | .340 | .375 | .410 | .460 | .510 | .620 | .700 | .76 |
| Friction easy bend | .080 | .110 | .130 | .148 | .165 | .180 | .207 | .225 | .275 | .310 | .343 |
| Area flue, sq. ft. for .03 in. frictional loss | 4.8 | 7.9 | 10.1 | 13.8 | 16.2 | 18.5 | 24.5 | 27.0 | 36.7 | 46.0 | 54.0 |
| Friction sharp bend | .123 | .170 | .210 | .235 | .265 | .290 | .330 | .362 | .430 | .490 | .542 |
| Friction easy bend | .054 | .077 | .092 | .105 | .113 | .128 | .153 | .160 | .197 | .222 | .240 |
| Area flue, sq. ft. for .01 in. frictional loss | 7.4 | 12.5 | 17.0 | 21.5 | 25.0 | 30.0 | 36.5 | 42.8 | 59.0 | 73.0 | 87.5 |
| Friction sharp bend | .053 | .070 | .087 | .104 | .110 | .125 | .135 | .148 | .180 | .205 | .228 |
| Friction easy bend | .023 | .032 | .039 | .045 | .048 | .053 | .060 | .065 | .079 | .091 | .100 |

frictional losses of 0.05 inch, 0.03 inch, and 0.01 inch of water for a flue 100 feet in length. The frictional loss in bends of a flue of the same sectional area is also given, the first case being for an abrupt right-angle bend, and the second case for an easy bend with the inside radius of the bend equal to the width of the flue.

The friction is based upon 500 cubic feet of gas per pound of coal, 4 pounds of coal per boiler horse-power, and the friction is given for iron flues of square section. For brick flues the

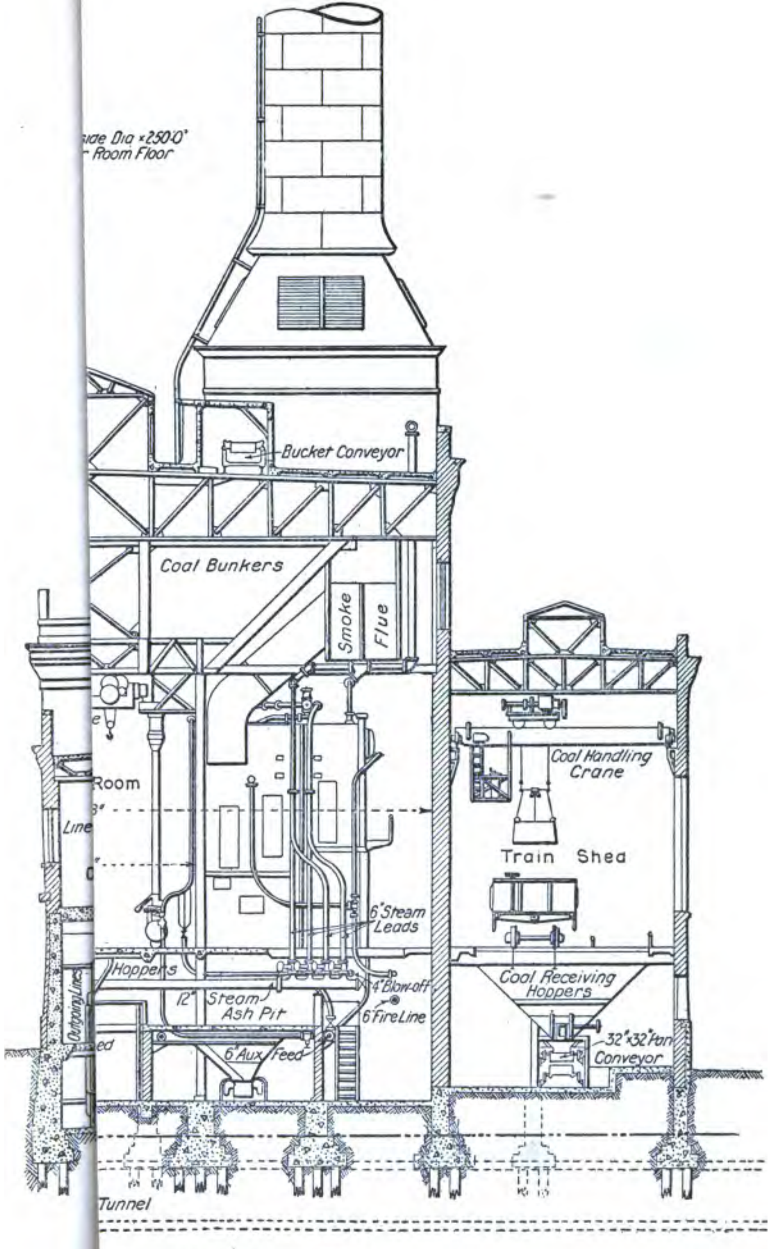
friction will be one-third greater, and for flues that are not square with sides as 1 : 2 the friction will be 7 per cent greater than that given. For round flues it will be 87 per cent of that given.

From the table it will be seen that a straight flue 100 feet long for 2500 horse-power of 46 square feet sectional area will have a frictional loss of 0.05 inch of water. A sharp bend in this flue will cause an additional friction of 0.76 inch of water and the sum of these will be too great for most cases, hence a flue of greater sectional area will be required.

Calculation has been made of the friction of gases in chimneys of the sizes given in the Kent table and the result is given in Table 33. The friction of each based upon 4 pounds of coal per horse-power hour and 500 cubic feet of gas per pound of coal, for both iron and brick stacks, is given. All of the data upon friction was obtained from a friction loss chart presented by Mr. Konrad Meier in his book on the "Mechanics of Heating and Ventilation," and based upon the following formula representing the frictional loss:

$$P_f = 0.075 \times 0.00624 \frac{V^{1.9}}{2g} l \left(\frac{c}{a}\right)^{1.18}.$$

The data upon friction may be used as follows: Suppose that it is desired to find the size of chimney and flue for a maximum capacity of 1000 boiler horse-power based upon the use of four pounds of No. 1 buckwheat coal per horse and that it is desired to burn the coal at the rate of 15 pounds per square foot of grate per hour, requiring a difference in pressure between the ashpit and furnace of 0.43 inch of water, the chimney and flue to be iron. It is further assumed that the load on the plant is fairly constant, that the boilers are to be run at their normal rating of 1000 horse-power, and that the flue-gas temperature will be 500 degrees and the outside-air temperature 60 degrees. The conditions are such that there will be a sharp right-angle bend in the flue as the gases enter the chimney, and the gases will have to make a similar turn in entering the flue from each boiler. From Table 31 it will be seen that the frictional loss of the gases passing through the fires will be 0.043 inch of water, and as this is quite high a tall chimney will be required, hence low losses in the flue are desirable. A flue of 42.8 square-feet area and 100 feet long and a chimney 60 inches in diameter and 200 feet high which is rated according to Table 33 as being of 1000 horse-power will be tried.




COMPANY, CHICAGO, ILL.

TABLE 33.—KENT'S TABLE OF CHIMNEY CAPACITIES BASED ON FOUR POUNDS OF COAL PER HORSE-POWER PER HOUR WITH FRICTION IN INCHES OF WATER.

| Diameter. | Area in square feet. | Height of chimney in feet. 50 | | | 70 | | | 100 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Friction. | | | Friction. | | | Friction. | |
| | | H.P. | Brick. | Iron. | H.P. | Brick. | Iron. | H.P. | Brick. | Iron. |
| 18 | 1.77 | 29 | .0130 | .0100 | 34 | .0258 | .0198 | | | |
| 21 | 2.41 | 44 | .0151 | .0116 | 51 | .0276 | .0212 | | | |
| 24 | 3.14 | 61 | .0146 | .0112 | 73 | .0289 | .0222 | | | |
| 27 | 3.98 | 81 | .0143 | .0110 | 90 | .0276 | .0212 | | | |
| 30 | 4.91 | 102 | .0138 | .0106 | 125 | .0268 | .0206 | 149 | .0511 | .0393 |
| 33 | 5.94 | .... | ...... | ..... | 156 | .0252 | .0194 | 186 | .0495 | .0381 |
| 36 | 7.07 | .... | ...... | ..... | 190 | .0233 | .0179 | 228 | .0460 | .0353 |
| 39 | 8.30 | .... | ...... | ..... | 227 | .0222 | .0171 | 223 | .0437 | .0336 |
| 42 | 9.62 | .... | ...... | ..... | 270 | .0213 | .0164 | 323 | .0421 | .0324 |
| 48 | 12.6 | .... | ...... | ..... | .... | ...... | ..... | 436 | .0389 | .0299 |
| 54 | 15.9 | .... | ...... | ..... | .... | ...... | ..... | 561 | .0350 | .0277 |
| 60 | 19.6 | .... | ...... | ..... | .... | ...... | ..... | 706 | .0333 | .0256 |
| 66 | 23.8 | .... | ...... | ..... | .... | ...... | ..... | 868 | .0298 | .0229 |
| 72 | 28.3 | .... | ...... | ..... | .... | ...... | ..... | 1045 | .0270 | .0208 |
| 78 | 33.2 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 84 | 38.5 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 90 | 44.2 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 96 | 50.3 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 102 | 56.8 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 108 | 63.6 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 114 | 70.9 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 120 | 78.5 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 132 | 93.0 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |
| 144 | 113.1 | .... | ...... | ..... | .... | ...... | ..... | .... | ...... | ..... |

| Diameter. | Area in square feet. | 125 | | | 150 | | | 175 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Friction. | | | Friction. | | | Friction. | |
| | | H.P. | Brick. | Iron. | H.P. | Brick. | Iron. | H.P. | Brick. | Iron. |
| 18 | 1.77 | | | | | | | | | |
| 21 | 2.41 | | | | | | | | | |
| 24 | 3.14 | | | | | | | | | |
| 27 | 3.98 | | | | | | | | | |
| 30 | 4.91 | | | | | | | | | |
| 33 | 5.94 | | | | | | | | | |
| 36 | 7.07 | 255 | .0710 | .0546 | | | | | | |
| 39 | 8.30 | 306 | .0694 | .0534 | 335 | .0990 | .0761 | | | |
| 42 | 9.62 | 361 | .0654 | .0503 | 395 | .0948 | .0729 | 427 | .1258 | .0966 |
| 48 | 12.6 | 486 | .0600 | .0461 | 532 | .0866 | .0666 | 575 | .1190 | .0915 |
| 54 | 15.9 | 628 | .0559 | .0430 | 689 | .0785 | .0603 | 744 | .1087 | .0836 |
| 60 | 19.6 | 790 | .0514 | .0395 | 845 | .0726 | .0559 | 935 | .0970 | .0746 |
| 66 | 23.8 | 970 | .0459 | .0353 | 1060 | .0655 | .0503 | 1149 | .0889 | .0684 |
| 72 | 28.3 | 1168 | .0408 | .0308 | 1280 | .0600 | .0461 | 1381 | .0801 | .0616 |
| 78 | 33.2 | 1380 | .0373 | .0281 | 1515 | .0557 | .0428 | 1638 | .0731 | .0562 |
| 84 | 38.5 | 1617 | .0354 | .0272 | 1770 | .0516 | .0397 | 1920 | .0687 | .0528 |
| 90 | 44.2 | 1870 | .0336 | .0258 | 2048 | .0497 | .0382 | 2215 | .0649 | .0499 |
| 96 | 50.3 | 2150 | .0320 | .0246 | 2342 | .0479 | .0366 | 2540 | .0612 | .0471 |
| 102 | 56.8 | 2430 | .0300 | .0231 | 2660 | .0456 | .0351 | 2878 | .0585 | .0450 |
| 108 | 63.6 | 2610 | .0271 | .0208 | 2990 | .0431 | .0332 | 3240 | .0544 | .0418 |
| 114 | 70.9 | .... | ...... | ..... | 3360 | .0386 | .0297 | 3630 | .0525 | .0403 |
| 120 | 78.5 | .... | ...... | ..... | 3738 | .0363 | .0279 | 4040 | .0506 | .0389 |
| 132 | 93.0 | .... | ...... | ..... | 4545 | .0347 | .0267 | 4920 | .0476 | .0366 |
| 144 | 113.1 | .... | ...... | ..... | 5450 | .0318 | .0245 | 5890 | .0419 | .0322 |

| Diameter. | Area in square feet. | 200 | | | 225 | | | 300 | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | Friction. | | | Friction. | | | Friction. | |
| | | H.P. | Brick. | Iron. | H.P. | Brick. | Iron. | H.P. | Brick. | Iron. |
| 18 | 1.77 | | | | | | | | | |
| 21 | 2.41 | | | | | | | | | |
| 24 | 3.14 | | | | | | | | | |
| 27 | 3.98 | | | | | | | | | |
| 30 | 4.91 | | | | | | | | | |
| 33 | 5.94 | | | | | | | | | |
| 36 | 7.07 | | | | | | | | | |
| 39 | 8.30 | | | | | | | | | |
| 42 | 9.62 | | | | | | | | | |
| 48 | 12.6 | 615 | .1522 | .1171 | | | | | | |
| 54 | 15.9 | 796 | .1394 | .1072 | 845 | .1712 | .1319 | | | |
| 60 | 19.6 | 1000 | .1262 | .0971 | 1060 | .1560 | .1200 | | | |
| 66 | 23.8 | 1228 | .1130 | .0870 | 1300 | .1443 | .1111 | 1500 | .2510 | .1930 |
| 72 | 28.3 | 1481 | .1050 | .0808 | 1570 | .1310 | .1009 | 1810 | .2358 | .1812 |
| 78 | 33.2 | 1750 | .0960 | .0738 | 1850 | .1176 | .0905 | 2145 | .2160 | .1660 |
| 84 | 38.5 | 2050 | .0879 | .0676 | 2170 | .1102 | .0848 | 2510 | .1925 | .1481 |
| 90 | 44.2 | 2370 | .0838 | .0644 | 2510 | .1064 | .0819 | 2895 | .1860 | .1431 |
| 96 | 50.3 | 2716 | .0794 | .0611 | 2878 | .1029 | .0791 | 3320 | .1781 | .1370 |
| 102 | 56.8 | 3090 | .0749 | .0576 | 3265 | .0963 | .0740 | 3760 | .1700 | .1308 |
| 108 | 63.6 | 3465 | .0708 | .0544 | 3670 | .0883 | .0679 | 4240 | .1625 | .1250 |
| 114 | 70.9 | 3880 | .0659 | .0506 | 4115 | .0844 | .0649 | 4750 | .1571 | .1210 |
| 120 | 78.5 | 4320 | .0636 | .0489 | 4570 | .0798 | .0613 | 5280 | .1489 | .1145 |
| 132 | 93.0 | 5250 | .0615 | .0472 | 5580 | .0765 | .0588 | 6440 | .1331 | .1023 |
| 144 | 113.1 | 6290 | .0536 | .0412 | 6660 | .0698 | .0536 | 7700 | .1179 | 0906 |

Before deciding upon the size the data given should be used to see if the draft from such a chimney will be sufficient. The various frictional losses in inches of water will be approximately as follows:

| | |
|---|---|
| Friction of fuel on grates | 0.430 |
| Friction of boilers | 0.200 |
| Friction of square flue | 0.010 |
| Friction of two sharp bends (2 by 0.148) | 0.296 |
| Friction of chimney | 0.097 |
| | 1.033 |

Now the theoretical draft for a 100-foot chimney with flue gas at 500 degrees and an outside temperature of 60 degrees is 0.639 inch of water, and for a chimney 200 feet high 1.278 inches, hence the chimney ought to be amply large enough to do this work. If the theoretical draft had not been sufficient the grate area might be increased to operate at a lower rate of combustion, which would reduce the frictional loss through the fires, or the chimney could be increased in height to increase the draft. The former is the cheaper method to pursue. While increasing the height of the chimney increases the theoretical draft directly as the height is increased, the friction of the chimney is increased at the same rate, but as the chimney friction, in this case 0.097 inch, is a small part of the total, this method of increasing the draft may be used.

**Capacity of Chimneys.** — Most chimney formulas are based on coal consumption but do not take into account the fact that the amount of gases given off by different coals varies. Col. E. D. Meier has called attention to this, and by calculating the volume of gases from their composition finds that the relative areas for chimneys for certain much-used coals should be as follows: Anthracite 100, New River (Va. semi-bit.) 93, Youghiogheny (Penn. bit.) 102, Mt. Olive (Ill. bit.) 128, Collinsville (Ill. bit.) 138.

Three well-known chimney formulas are those of Kent, Gale, and Christie. In the first

$$\text{H.P.} = 3.33\, E \sqrt{H},$$

in which $H$ is the height in feet and $E$ is the effective area, the actual area being determined by increasing the diameter of the chimney if it be round or the side of the chimney if it be square

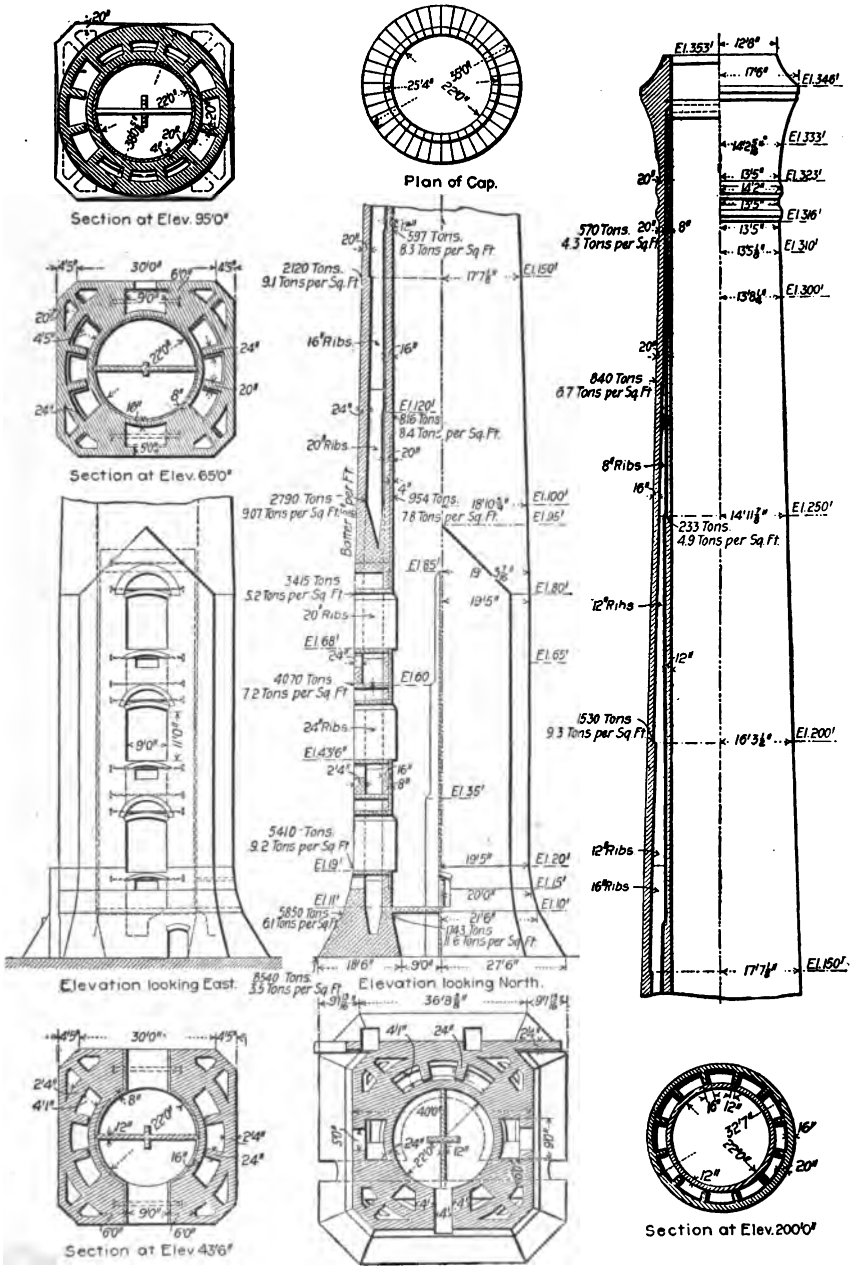


Fig. 54. Chimney, Metropolitan Street Railway Co., New York.

by four inches to allow for the lining of the chimney by a layer of gas that is assumed to have no velocity. $E$ is approximately equal to $(A - 0.6\sqrt{A})$ for round chimneys and to $A - \frac{2}{3}\sqrt{A}$ for square chimneys. As this formula is based upon the burning of five pounds of coal per horse-power

$$C = 16.6 E \sqrt{H},$$

where $C$ equals the number of pounds of coal burned per hour. The Kent formula assumes that the height and area are interdependent, and it only holds within certain limits. Kent's table of chimney capacities is given in Table 33 with the frictional loss in inches of water.

Gale's formula may be expressed in the form:

$$A = 0.07\, C^{\frac{1}{2}} \text{ and } H = \frac{180}{t}\left(\frac{C}{G}\right)^2,$$

in which $t$ is the temperature of the chimney gases and $G$ the grate area in square feet. Col. E. D. Meier, who has a great deal of experience with western coals, states that this formula gives rather too large results and recommends that:

$$H = \frac{120}{t}\left(\frac{C}{G}\right)^2,$$

and that after the height is found by this formula the area be obtained by the one proposed by Mr. Kent.

Mr. George A. Orrok in "*Power*" states that the constant in the general chimney formula, for which Kent gives the value of 16.66, varies greatly in the formulas of different authorities, and Mr. Orrok recommends that a value of 12 be given it for brick-lined stacks, but that in case of an unlined steel stack the value of this constant may be increased to 14 or 15 and for small stacks 16 may be used.

Mr. W. W. Christie in his book on "Chimney Design" gives the chimney formula: Horse-power $= 3.24\, A\sqrt{H}$, it being assumed that four pounds of coal are burned per horse-power, $A$ being the area of the flue and $H$ the height, both in feet. If $C$ is the coal burned per hour $C = 12.96\, A\sqrt{H}$.

**Thickness of Chimney Walls.** — In designing a chimney of a given height and inside diameter it is necessary to determine the

thickness of the walls required to provide sufficient weight to prevent its being overthrown by the wind. It is customary to step out the inner walls at different levels so as to divide the shell into a series of sections, each of a uniform thickness which is less than that of the section immediately below it. As the walls have a slight batter inside and outside, the diameters at the top and bottom and the thickness and heights of the different sections of the shell have to be determined. To simplify the problem it is proposed to give a method of designing a chimney with straight inner and outer walls of the proper batter. From this a chimney of approximately equivalent weight but made up of several sections each of uniform thickness can be designed.

It is usually customary to make the weight of a chimney such that it will bear a certain relation to the wind pressure, a rule commonly used requiring that the prolongation of the resultant of the total wind pressure acting through the center pressure and the weight of the chimney intersect a horizontal plane through the chimney base at a point not more remote from the axis of the chimney than a distance equal to $D_b \div 6$ where $D_b$ represents the outside diameter at the base, or at the elevation at which the calculation for stability is made. If the chimney be round, square, or octagonal, the diameters referred to are those of the inscribed circles. To illustrate, if, in Fig. 56, $CD$ represents the wind pressure considered to be acting through the center of pressure $C$, and $CE$ represents the weight of the chimney, and the prolongation of their resultant $CF$ intersects the base $AB$ at a point $G$ distant from $H$ by a distance less than $D_b \div 6$, then the conditions as to stability are fulfilled. In Fig. 56, $C$ is the center of pressure; $CD$ represents the total wind pressure $P$; $CE$ the weight $W$; and $CH$ equals $h$, the height in feet of the center of pressure above the base $AB$. Expressed algebraically the formula is:

$$\frac{W}{P} = \frac{h}{D_b \div 6}. \tag{1}$$

The value of $W$ can be found from the volume of the chimney, assuming that one cubic foot of brick masonry weighs 115 pounds. Then $W = 115\ V$. Therefore:

$$W = 115\, H\pi\left(\frac{(D_b^2 + D_t^2 + D_bD_t) - (d_b^2 + d_t^2 + d_bd_t)}{12}\right). \tag{2}$$

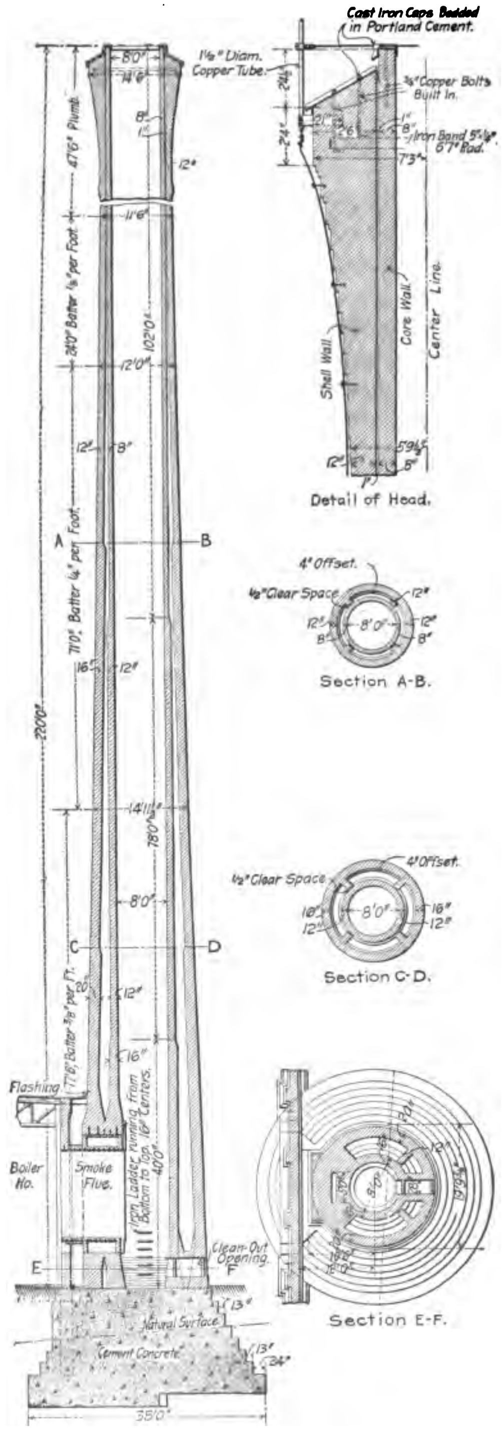


Fig. 55. Brick Chimney designed by Lockwood, Greene and Co.

Throughout this discussion the terms used have the following significance:

$H$ = height of chimney above base in feet.
$h$ = height of center of pressure above base in feet.
$D_t$ = outside diameter top in feet.
$d_t$ = inside diameter top in feet.
$D_b$ = outside diameter bottom in feet
$d_b$ = inside diameter bottom in feet.
$P$ = equivalent total wind pressure.

Now if the value of $W$ in equation (2) be substituted for $W$ in equation (1) then:

$$115\pi H\left(\frac{(D_b^2 + D_t^2 + D_bD_t) - (d_b^2 + d_t^2 + d_bd_t)}{12}\right) = \frac{6\,hP}{D_b}. \quad (3)$$

There is but one factor in this equation, $d_b$, whose numerical value is not known either from data assumed at the outset or from its relation to known quantities in the equation. This being the case the equation can be solved for the value of $d_b$ and its numerical value obtained. One would then have all the dimensions of a chimney with straight walls of a height and inside diameter necessary, and which would satisfy conditions of stability.

The inside diameter of the chimney at the top and the height must be assumed. From the former dimension the outside diameter at the top and the outside diameter at the bottom can be found readily by two empirical rules. The first of these is Professor Lang's rule, given in his paper on the "Construction and Dimensions of Chimneys for Boiler Plants," of which a translation was printed in *The Engineering Record* of July 20 and 27, 1901. It is to the effect that the thickness in feet of a chimney at the top, $t$, neglecting the ornamentation, should be:

$$t = 0.328 + 0.05\,d_t + 0.0005\,H;$$

therefore:

$$D_t = d_t + 2\,t = 0.656 + 1.1\,d_t + 0.001\,H. \quad (4)$$

The minimum value of $t$ should be 0.58 foot for radial brick and 0.7 foot for common brick. As the thickness must be over a brick's length and as it cannot increase by less than half a brick's length, the value of $t$ must be 0.7, 1.08 feet, etc. Few

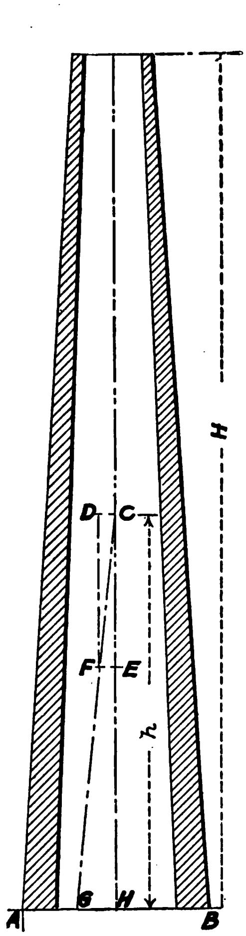



Fig. 56.

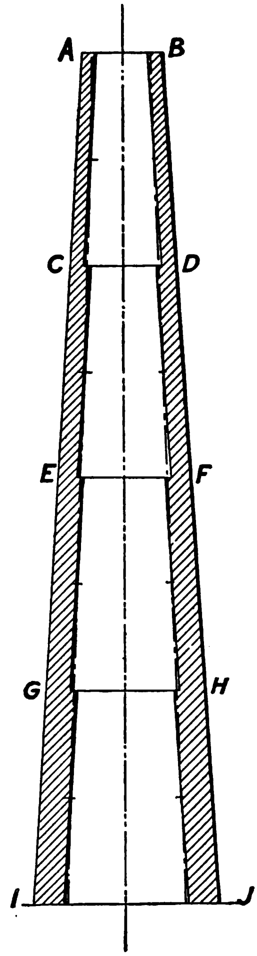



Fig. 57.

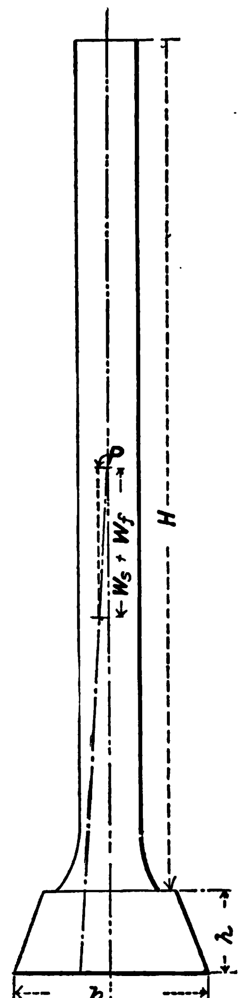
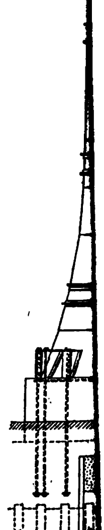



Fig. 58.

chimneys built of common brick have a value for $t$ greater than 1.08 feet. The outside diameter at the bottom, or $D_b$, can be obtained from the outside diameter at the top by a rule which assumes a batter for the outside wall of 1:30 to 1:36 on a side. Assuming it to be 1:32 then:

$$D_b = D_t + \frac{H}{16};$$



substituting the value of $D_t$ from equation (4) then:

$$D_b = 0.656 + 1.1\,d_t + 0.001\,H + \frac{H}{16} = 0.656 + 1.1\,d_t + 0.063\,H.$$

The value for $P$, the total wind pressure in equation (3), is found by multiplying the assumed wind pressure of 50 pounds per square foot by the equivalent of the vertical cross section of the chimney through the axis, in square feet. With a square chimney this plane is taken parallel to two opposite sides. For square chimneys $P$ has a value equal to their cross section in square feet, for round chimneys 0.50 times this area, and for octagonal ones 0.71 times this area.

The numerical values of $H$, $D_b$, $D_t$, $d_t$, $P$, and $H$ found in the manner indicated are substituted in the above equation, which then can be solved for the value of $d_b$. This determined we have all the dimensions of the straight-walled chimney of the height and inside diameter required for the capacity, and which fulfills the conditions as to stability. As has been said, the interior of a chimney is seldom constructed with a straight batter, but in a series of steps each section from the top downward increasing in thickness by half a brick's length. Each step should then be of a thickness in inches that is divisible by $4\frac{1}{2}$, this being the width of a brick and the necessary mortar. From the top down, therefore, the thicknesses of the different sections should be $8\frac{1}{2}$, 13, $17\frac{1}{2}$, 22, $26\frac{1}{2}$, etc., inches. Should the calculations for the thickness of the wall at the bottom call for a thickness intermediate between two of these thicknesses, the greater one can be selected as the thickness of the bottom section. Should such calculation show that the thickness at the bottom should be 22 inches, the chimney can be divided into four sections of equal height, $8\frac{1}{2}$, 13, $17\frac{1}{2}$, and 22 inches thick. Radial brick for chimneys are made in several sizes so that the thickness, when they are used, increases by about two inches at the offsets.

After the thickness of walls of the different sections and their heights are determined, the calculation for stability must be made at the base of each section, paying no attention to that below. For instance in Fig. 57, the stability of the part $ABJI$ should be calculated considering $IJ$ as the base, also $ABHG$ considering $GH$ the base, etc., each time locating the center of pressure for the part under consideration.

Another operation yet remains, and that is to find the pressure imposed upon the brickwork by its own weight. Calculation must be made as to the pressure per square inch at the base, and at each level where the shell changes in thickness, that is, at the bottom of each section. According to Professor Lang, the pressure in pounds per square inch should nowhere exceed that given by the formula $P = 71 + 0.65\,L$, where $L$ denotes the distance in feet from the top of the chimney to the point in question. Should the pressure exceed that given by the formula the walls of the chimney should be made thicker. The function of the chimney height is introduced in the formula to allow a greater pressure in high chimneys, which are erected less quickly than shorter ones, the mortar therefore having more time to harden.

**Chimney Linings, etc.** — Chimneys of common brick built in the United States are usually provided with an inner core or lining to protect the outer shell from the heat. Sometimes this core is carried up for half the height of the chimney, but more usually to the top. Care should be taken that it is built independently of the outer shell as the greater expansion of the core would injure the shell. With radial brick chimneys the inner core is not so common as the bricks are more carefully burned and selected than is the custom with common brick and are not so easily affected by heat. Any chimney likely to contain gases at a higher temperature than 600° F. should be lined with fire brick set in fire clay or lime mortar, preferably the former. Sometimes the fire-brick lining extends from the bottom to one-third or one-half of the height. The core can be divided into sections, each about 40 to 50 feet high, and 4, 8½, 13, 17½, and 21 inches in thickness from the top down. If the chimney is of fairly large diameter the 4-inch section should not be over 25 or 30 feet. In the largest chimneys 8 inches is the minimum thickness of the core. It usually has a uniform inside diameter so that changes in thickness are secured by offsets on the outside. With the batter for the outer surface of the outside shell recommended in an earlier paragraph there is likely to be sufficient distance between the offsets of the core and those of the outer shell to provide proper clearance. When the chimney is drawn on paper this can be determined. Steel chimneys should also be lined to prevent loss of heat and also air leakage, which will

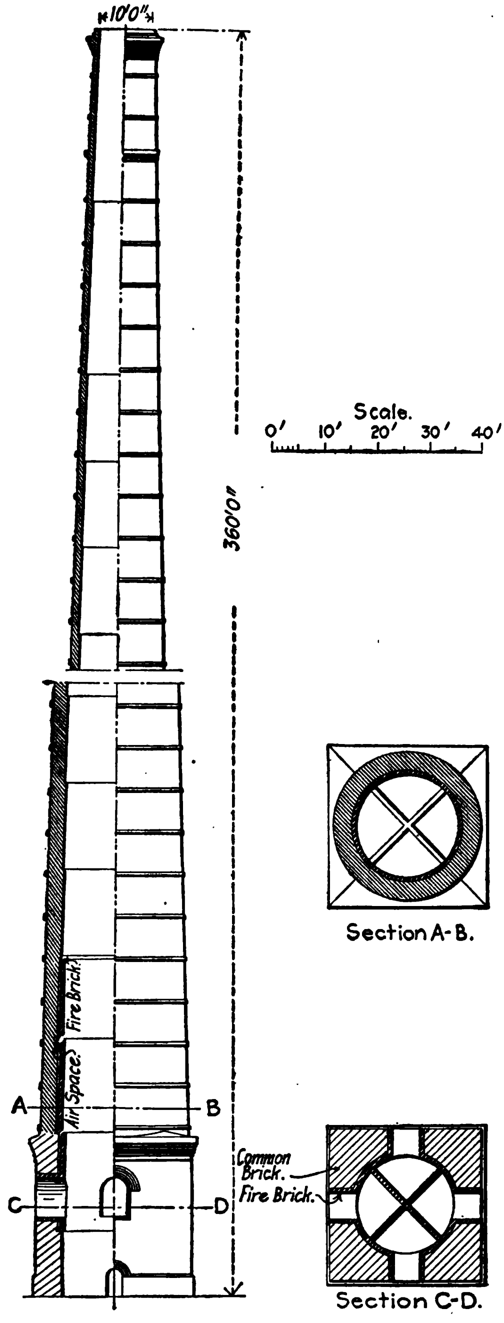



Fig. 59. Custodis Radial Brick Chimney for Orford Copper Co.

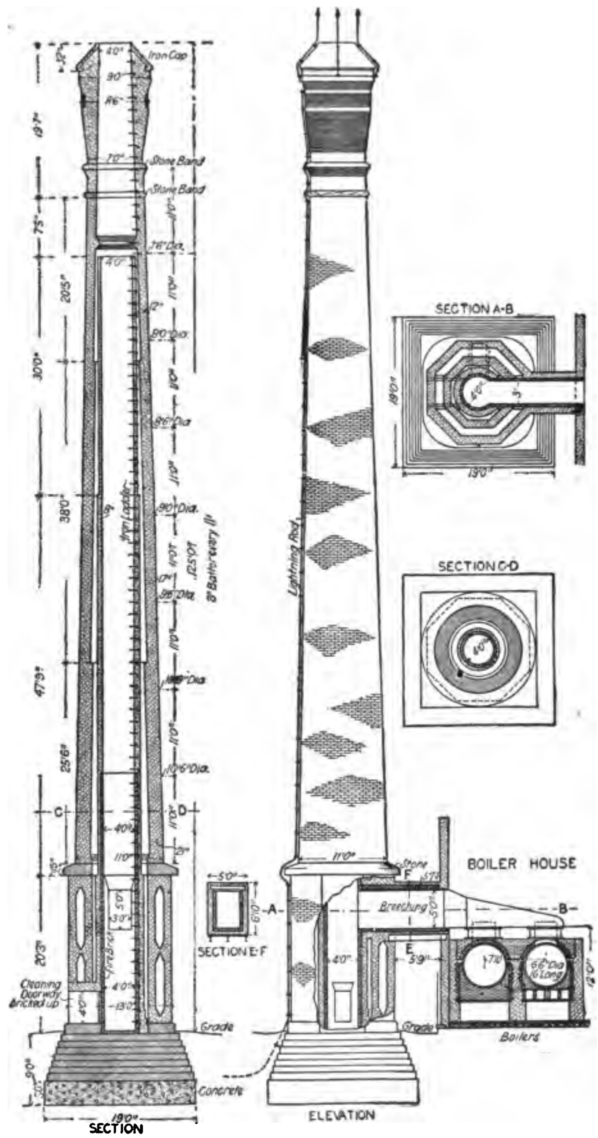



Fig. 60. Chimney, Laidlaw-Dunn-Gordon Co., Cincinnati, Ohio.

occur unless the joints are carefully calked, a provision that is frequently overlooked.

In constructing the opening for smoke flues at the base of the chimney care should be taken that it is not weakened at that point. The top of the opening should be arched over or spanned with heavy steel beams built in the masonry. Should the chimney have an inner core the smoke flue should, of course, be continuous through both shells. Ladders for reaching the top of a chimney are usually located on the inside of brick chimneys and more frequently on the outside of steel ones. If the latter, the continuous hand rails at the sides should be so designed as to allow for the possibility of a difference in expansion between the ladder and the chimney. The tops of brick chimneys should be covered with a cast-iron cap held in place by anchor bolts, and lightning rods metallically connected to the ground should also be provided.

**Materials.** — Masonry chimneys are usually built of brick, but recently concrete chimneys reënforced with steel, as in the Ransome chimney, shown in Fig. 61, have been used with success. For brick chimneys hard burned brick of high specific gravity should be used. A special brick for chimney building used to a large extent in Europe and to a rapidly increasing extent in the United States is the radial brick for round chimneys, with its inner and outer surface curved to conform to the curvature of the chimney. Several holes running through the brick aid in the burning and also serve to secure it in place more strongly as the mortar works into them when the brick is laid. The radial brick that have been used to a considerable extent by the Custodis and Heinicke companies in Germany are much stronger and more durable than common brick. Another feature in their favor is that they are considerably larger than common brick and less labor is required in laying them.

**Masonry Chimney Foundations.** — These should be of such an area that the load per square foot does not exceed one ton per square foot on soft clay; two tons per square foot on stiff clay, compact sand, loam, etc. These loads are exceeded in buildings but they should not be in chimneys unless solid rock underlies the foundation. The rock should be dressed off into steps with vertical sides so there will be no tendency to slide. With very large masonry chimneys, and in fact with chimneys

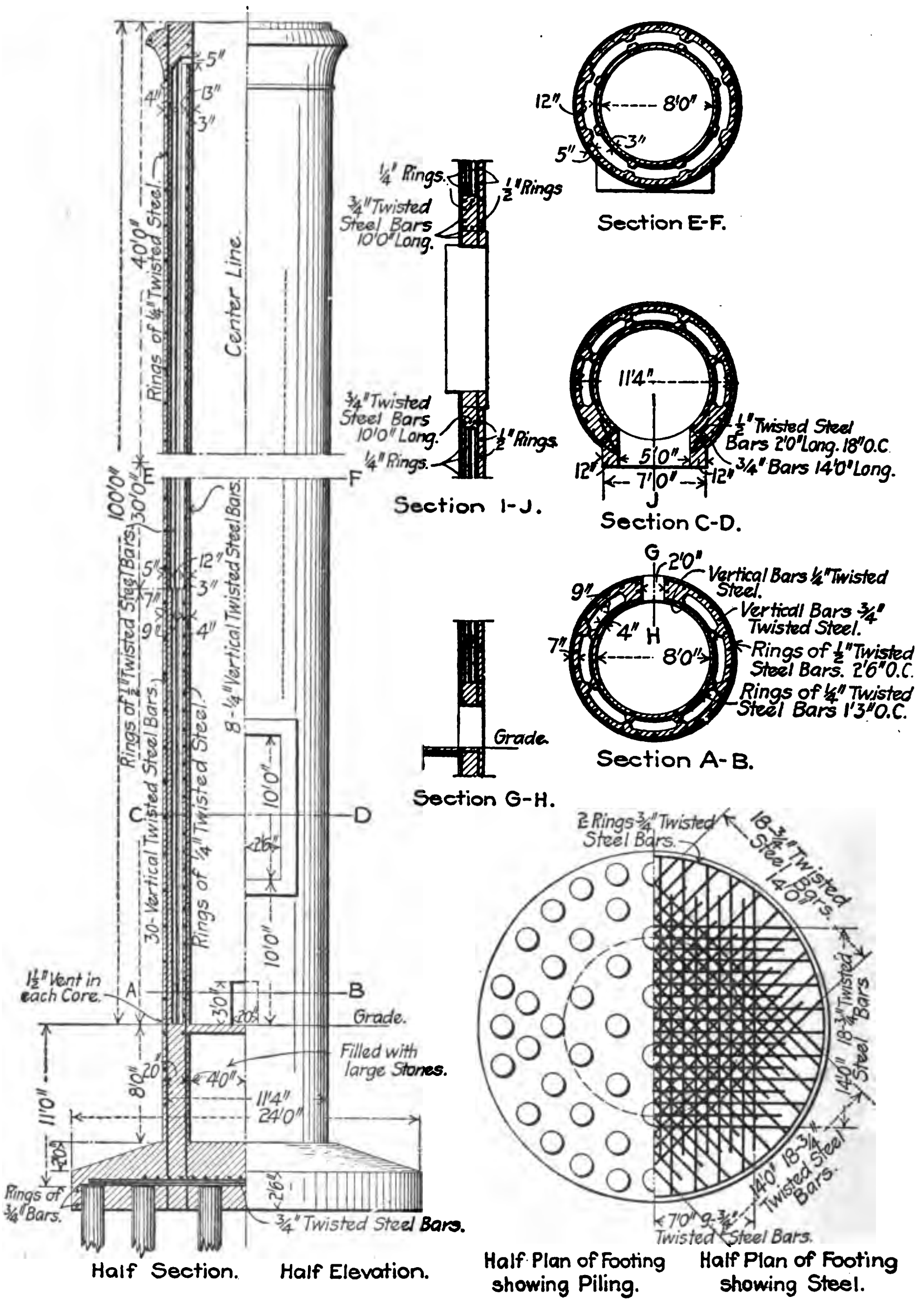



Fig. 61. Ransome Concrete Steel Chimney, Central Lard Co., Hoboken, N. J.

of moderate size in soil of low-bearing power, pile foundations are frequently resorted to, the piles being driven on about $2\frac{1}{2}$-foot centers and cut off below the level of surface water; they support a concrete bed two or more feet in thickness into which their tops extend. Concrete or brick foundations laid in cement mortar should be laid several weeks before the chimney is constructed in order that the cement should set properly. The sides of the foundations which are usually in the form of a truncated pyramid should have an inclination of at least 60 degrees to the horizontal. The depth of the foundation should be such as to properly distribute the load.

**Steel Chimneys.** — Chimneys built of sheet steel are common, particularly in Pennsylvania and the middle West. They may be either self-supporting or held in position by means of guy ropes. To save expense, it is not unusual for a considerable number of small guyed chimneys to be used in place of one large chimney. Steel chimneys are said to cost less than those built of brick but the market price of steel affects their relative cost considerably. Any steel chimney should be carefully calked at the joints and the vertical lap joints scarfed at the girth seams. The leakage of air that would otherwise occur and which unfortunately does occur in many cheaply built steel chimneys very greatly affects the draft. Self-supporting steel chimneys rest usually on cast-iron base plates sometimes cast in sections and bolted together if the chimney be large enough to make this desirable. The base plate is held down by foundation bolts built in a brick or concrete foundation sufficiently heavy to prevent the chimney from overturning. The lower courses of a self-supporting stack are usually flared out at the base of the chimney in a conical or bell shape, to give stiffness to it, the height of this cone or bell being from $1\frac{1}{2}$ to 2 times the diameter of the chimney above the bell and of a diameter at the base equal to the height of the bell.

A formula for determining the thickness of shell that is used by a firm which has built a large number of large self-supporting steel chimneys is as follows:

$$\frac{\text{moment in inch pounds}}{0.7845\, D^2} = \text{stress per lineal inch.}$$

This assumes that the moment of the total wind pressure in pounds multiplied by the distance in inches of the section under

consideration from the center of pressure, divided by the diameter of the chimney in inches squared multiplied by 0.7854, is equal to the maximum stress per lineal inch in the shell. The total wind pressure is based on an assumed pressure of 25 pounds per square foot of projected area. A safe working stress is 10,000 pounds per square inch and this should be reduced by the efficiency of the riveted joint. If the efficiency of the riveted joint is 60 per cent, then 6000 pounds per lineal inch would be a safe working strength and the ratio of the stress per lineal inch, as found by the equation, to 6000 would be the thickness in inches of the shell required at the section under consideration. Calculation for the thickness of the shell should be made at the base of the stack, the top of the bell, and several points between it and the top. The greatest strain occurs at the top of the bell. On account of deterioration that is apt to occur in the steel, it is undesirable to use shells less than $\frac{3}{8}$ to $\frac{1}{4}$ inch in thickness, depending upon the size of the chimney. This is particularly true near the top where the greatest corrosion is apt to occur, owing to the effect of the smoke that usually clings to the lee-side of a chimney.

For about one-fifth or one-quarter the height of a self-supporting chimney it is desirable that the girth seams be double-riveted. The lower edge of any sheet usually overlaps the upper edge of the sheet beneath it. For the sake of the greater stiffness, the vertical seams at the bell should also be double-riveted.

**Foundations for Self-Supporting Steel Chimneys.**—The foundations for self-supporting steel chimneys should have such a base that the load caused by the weight of the chimney and base and by the wind pressure on the leeward half of the base does not exceed the requirements of safety. The weight of the lining, if there be any, is to be neglected. The foundation must have sufficient mass so that the moment of the wind pressure by the height of the foundation plus half the height of the steel shell shall be equal to the weight of the shell plus the weight of the foundation multiplied by one-third the length of the base of the foundation.

If $P$ is the total wind pressure, $W_s$ the weight of the stack, $W_f$ the weight of the foundation, all in pounds, then in Fig. 58 the conditions as to stability are fulfilled when

$$P\left(\frac{H}{2} + h\right) = (W_s + W_f)\frac{b}{3}.$$

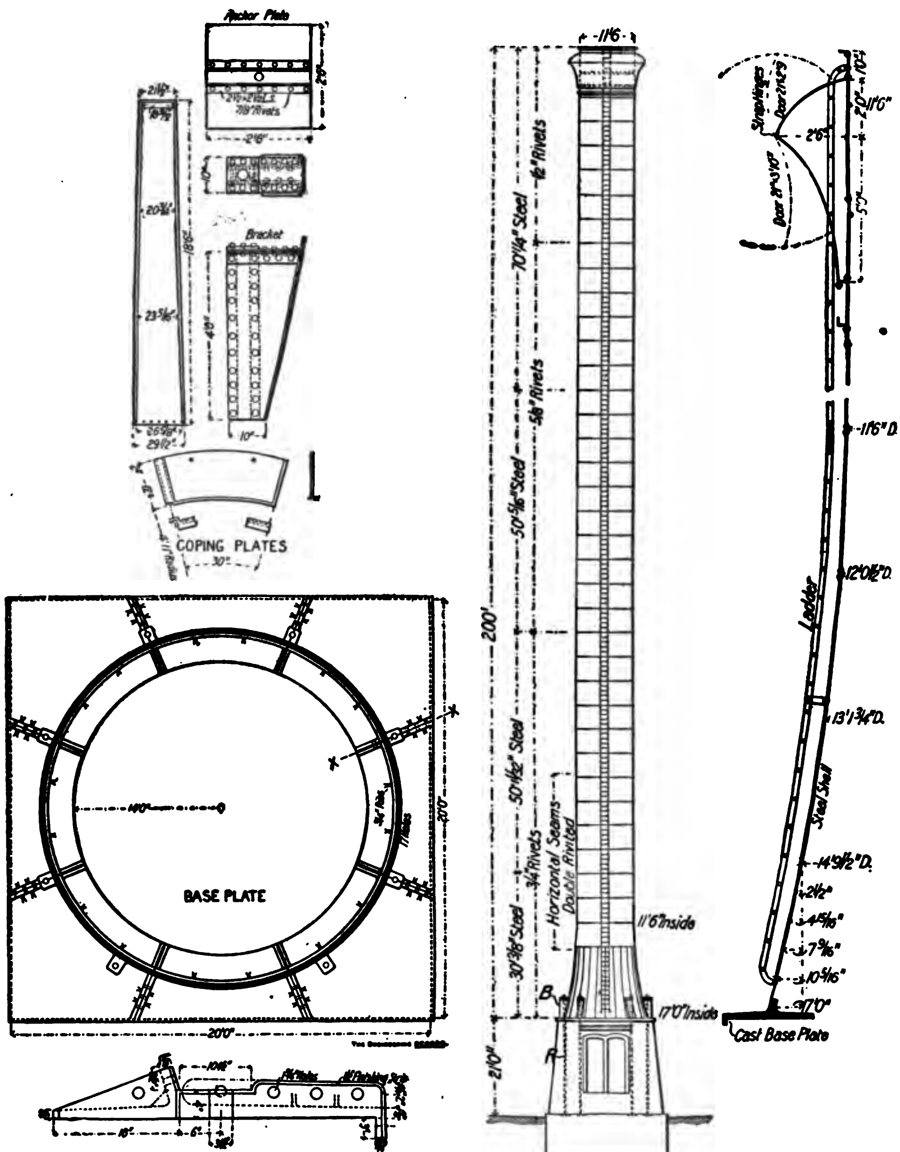



Fig. 62. Steel Chimney at Wilmerding, Pa. (Built by the Riter-Coaley Manufacturing Co.)

The wind pressure can be obtained by calculation and the height of the foundation can be taken at from one-eighth to one-tenth the total height of the chimney. With these data and the weight of the stack being known, the weight of the foundation can be calculated. As concrete weighs about 140 pounds per cubic foot, the volume of the foundation can be determined and from this the area of the top and bottom. The slope of the sides can be to 1 to 3. Should the wind pressure be found to cause too great a load along the leeward edge of the foundations, a foundation of less depth which would insure one of greater area can be assumed.

Information as to the size of smoke flues is given in the article on "Draft" in this chapter. Material for flues may be of masonry, preferably brick, of black iron varying from No. 10 for very small boilers up to, say, $\frac{3}{8}$ inch thick for large plants. For substantial work with a smoke flue 4 feet by 5 feet in section $\frac{3}{16}$-inch iron may be used with $2\frac{1}{2}$ by $2\frac{1}{2}$-inch angles at all corners, and at circumferential joints with stiffening angles of the same size on top placed about four feet apart. Expansion joints of the sleeve type should be placed in long flues when expansion might be troublesome. The main damper should have outside lever for damper connection with checks limiting the swing through an angle of 90 degrees. Clean-out doors should have bar iron frames, latch, and hinges. Dampers in branch connections from main flue to each boiler should have some form of locking device so they may be fixed in any desired position.

# CHAPTER XIII.

## COAL HANDLING, WATER SUPPLY, AND PURIFICATION.

**Coal-handling Machinery.** — One of the most important factors governing the selection of a location for a steam power plant is the cost at which coal, assuming this fuel is to be used, can be transported to the boiler plant and the cost of the disposal of the ashes. Because coal can frequently be conveyed more cheaply by water than by rail large power plants are located upon navigable waterways, if it is convenient to do so, and if not, the site should be near a railroad if possible, so that coal can be delivered at the minimum expense. Of course the expenditure to which one should go in providing means for handling coal and ashes depends entirely upon the amount to be burned, as the cost of handling coal may be so small compared to other expenses as to be of secondary consideration. Even in the smallest plants, however, thought should be given the matter of coal delivery. If coal-handling machinery is not used it is a convenient arrangement to have coal pockets or coal bunkers located next to the wall of a boiler room so that coal will fall from them by gravity to the floor in front of the furnace doors. If possible, arrangements should be made for delivering the coal to the bunkers directly from the coal cars, whether they be steam-railroad coal cars or small dumping cars of an automatic railway run from some near-by wharf. As plants increase in size greater expense for coal-handling apparatus is warranted. Means should be provided for storing coal and the amount of storage space required depends, in a measure, upon the effect of an enforced shutdown. For an electric-lighting or railway power station, or for power and heat for hospital buildings or public institutions, large storage capacity is imperative, particularly in localities where there is likely to be an interruption in the supply of fuel.

With plants as small as 500 or 600 horse-power in boilers, coal-handling machinery is frequently installed. It usually consists of a receiving hopper, into which the coal is delivered, feeding to an endless chain of buckets which elevates and conveys the coal to bunkers placed over the boilers; spouts lead from the bunkers to the floor near the furnace doors of the boilers if they are hand-fired or to mechanical stokers if the latter

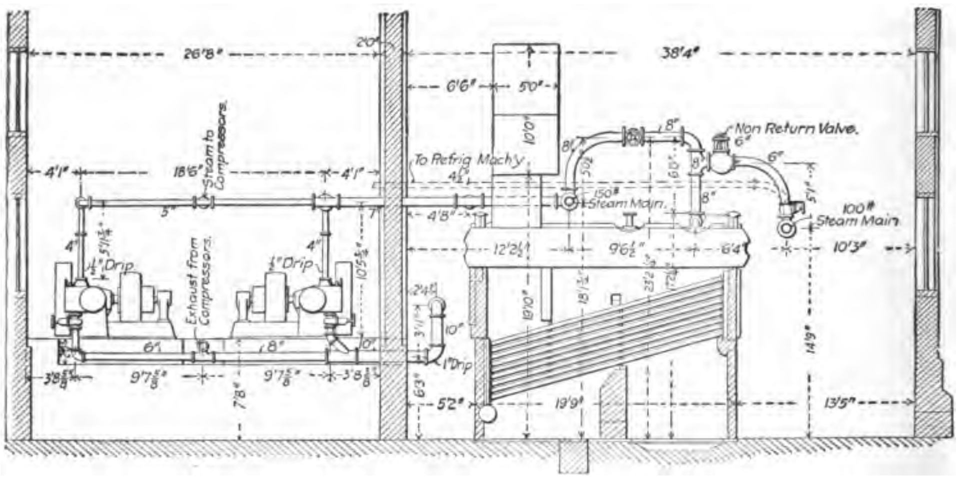



Fig. 63. Cross Section, Power House Central Lard Co.

are provided. Frequently the conveyor is arranged to pass beneath ash hoppers under the boiler grates so that when not handling coal it can be used to elevate the ashes into a storage bin, from which they can be removed by carts or otherwise. When coal is delivered by boat to a plant, particularly in large stations, it is frequently elevated by a self-filling bucket and raised by a boom to an elevated coal hopper in a tower on the wharf; the coal falls through a crusher to reduce it in size and then passes through automatic weighing scales to the conveyors which deliver it into bunkers over the boilers. Sometimes large storage bunkers are built outside of a power house, these being filled by conveyors, while a second conveyor extends from the storage bunker to a smaller bunker over the boilers. Some idea of the varied manner in which coal-handling machinery for power houses may be applied can be had by examining the illustrated catalogues of makers of this class of machinery.

**Cost of Coal Handling by Machinery.** — For the purpose of showing the relative cost of hand firing and a modern coal-handling equipment combined with mechanical stokers in large plants, some pertinent figures have been obtained through the courtesy of a well-known electrical-supply company owning a plant, containing about 7500 horse-power in boilers, which was operated for some time after construction without any kind of coal-handling machinery other than small hand cars which were loaded by hand from railway cars outside of the building and then hauled up a slight incline to the boiler house, so that the fuel could be dumped in front of the furnaces. The cost is given for two months, a year apart; during the first month, the plant was run without and during the second month with coal-handling machinery and mechanical stokers. The coal-handling plant consisted of a McCaslin conveyor so arranged that the coal was only handled by hand in shoveling it out of railway cars onto the conveying system:

| May, 1900. | Wages. | Tons of coal burned. | Cost per ton. |
|---|---|---|---|
| 16 firemen and one helper | $981.80 | 4292 | $0.229 |
| 11 coal and ash men. Ash removing by contract | 634.66 | ......... | 0.1478 |
| May, 1901. | | | |
| 3 firemen and 2 helpers | 287.75 | 6975 | 0.041 |
| 11 coal and ash men, 2 conveyor men | 654.50 | ......... | 0.0938 |

The saving in wages of firemen and helpers amounts to 18.8 cents per ton, which is 82.1 per cent or $1311.30 per month. The saving on coal and ash handling is 5.4 cents per ton, which is 41.4 per cent or $376.55 per month, or a total saving of $1687.85 per month or over $20,000 per year. Were it not for the fact that, owing to peculiar local conditions, the coal has to be shoveled from the coal cars onto the conveyor system the cost of labor might be still further reduced.

**Cost of Boiler-room Labor.** — This matter is of importance in deciding upon methods of handling coal in steam power plants, and some valuable information in this connection was contained in a report by Mr. R. S. Hale to the Steam Users' Association, whose members represented nearly 400 mill owners, largely in New England and the Middle States. Judging from the replies

received from members owning a total of about 600 boilers, it costs to move coal by hand (wheelbarrow) about 1.6 cents per ton per yard up to distances of five yards, then about 0.1 cent

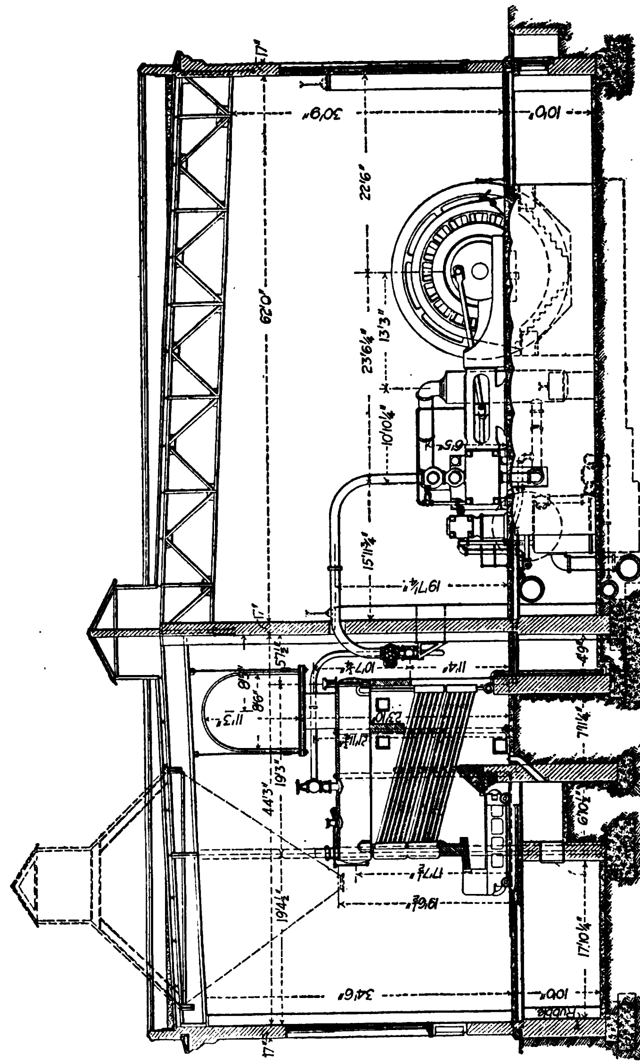


Fig. 64. Cross Section, Power House Des Moines City Railway Co. (Sargent and Lundy, Engineers.)

per ton per yard for each additional yard. Mr. Hale found that one man, besides a night man, can run an engine and fire up to about 10 tons of coal per week. One man, besides an engineer

and night man, can fire up to about 35 tons per week. Two men, besides an engineer and night man, can fire up to about 80 tons per week. These figures assume that the night man does all he can of the banking, cleaning, and starting. The

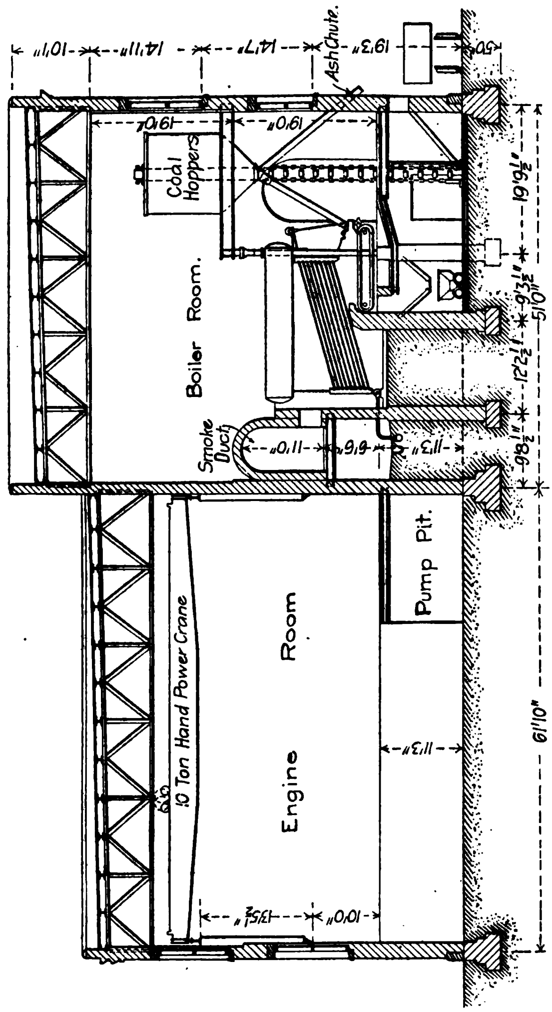


Fig. 65. Cross Section, Power House Schwarzchild and Sulzberger, Chicago, Ill. (L. Levy, Chief Engineer.)

figures are for average conditions. If the conditions are exceptional, as, for instance, where there is a very long wheeling distance or a very variable load, proper allowance should be made. Mr. Hale states in the report that mechanical stokers save from

30 to 40 per cent of the labor in plants burning from 50 to 150 tons per week and save no labor in small plants. Boiler attendants are paid about $1.50 per day, working from 10 to 12 hours. The average cost of firing coal, according to the report, was 48 cents per ton, the maximum 71 cents and the minimum 26 cents.

**Mechanical Stokers.** — In the chapter on mechanical draft it was explained that perfect combustion required just the proper amount of air to be supplied the furnace of a steam boiler. That requirement is difficult to fulfill where fuel is fired by hand. If the bed of fuel is too thin there will be an excess of air, and if too thick too little air will enter the furnace, both of which will cause a loss for reasons previously explained. This is particularly true with bituminous coals. Theoretically, therefore, the best result would be obtained by firing small quantities at frequent intervals rather than larger quantities of fuel less often. Frequent firing requires that the fire doors be opened often, and this means a loss due to too much air entering while the doors are open. The practical objection to frequent firing lies in the fact that it is difficult to get a fireman who will fire a boiler properly, most of the men who perform this kind of labor being of the class that prefers shoveling a large amount of fuel into the furnace and then sitting down for half an hour than to shoveling a lesser amount at shorter intervals. The kind of men who take an interest in stoking properly, and do it, are not likely to remain firemen. For the reasons given, the only way in which coal can be supplied to a furnace in such a manner as to produce the best results is by mechanical means or mechanical stokers. There is every reason to believe that a plant of boilers using bituminous coal mechanically fired will operate with a sufficient economy of fuel over hand-fired boilers to pay a good return on the increased investment required for the stoking apparatus. This result can be effected by any fireman who is sufficiently intelligent to do ordinary firing but not, however, unless the mechanical stoker is given some attention to see that holes in the fire do not occur or that clinker does not form to impede the uniform movement of the coal. The stoker is not wholly automatic, but, given a fair degree of attention, it is an indispensable aid to the firemen in the attainment of the perfect combustion of bituminous coal. Mechanical stokers are also of value in

reducing smoke with bituminous coal and reducing the number of men in the boiler plants. They are of particular value in making it possible to burn low-grade smoking coals in situations where the emission of smoke would be objectionable. As an improper air supply affects the amount of smoke emitted by coal, mechanical stokers properly designed very much reduce the smoke and oftentimes entirely do away with it.

**Burning Pulverized Fuel.** — There is one method of mechanically supplying coal to boilers that gives promise of much success and that is after pulverization. The principal difficulty that has been met in the past has been the expense of pulverizing, but recently methods have come into use that make this method commercially possible. Various types of mills for this purpose have been in use in the cement industry, ever since the development of the rotary kiln, that have given excellent satisfaction. Bituminous coal is ground and stored in hoppers from which it is delivered usually by a small screw conveyor to a pipe about four inches in diameter and terminating in the mouth of the kiln in which the cement is burned. A blast of air passing through the pipe delivers the coal to the kiln where it burns. The jet of flame is six or eight feet long and of such an intensity that it could not be introduced directly under a steam boiler, hence a boiler would have to be equipped with a detached or separate furnace to use fuel in this way. One difficulty with this method of using pulverized fuel lies in the danger of storing it in quantities, and to overcome this machines have been devised to pulverize the coal and deliver it directly to the furnace, each boiler being equipped with a pulverizer. By pulverizing the fuel the combustion of all the coal is complete as there is no loss due to the falling of fine particles of coal through the grate bars as there is in other methods of firing. The relative amounts of air and coal supplied can be adjusted to a nicety, hence there is no loss through too little or too much air. Furthermore, the coal can be burned with absolutely no smoke.

**Supply of Boiler Water.** — The amount of water used for steam boilers in large plants is of such a quantity that an abundant supply of good water at low cost is a deciding factor in the selection of a site for a power house. The cost of water from city mains is usually such that it is desirable for large plants to go to some other source, as a river, if one be available, artesian wells,

etc. Before determining upon a supply, investigation should be made as to whether or not the water is suitable for boiler purposes, the opinion of a chemist, not a boiler-compound quack, being obtained upon this point. Water may be unfit for boiler purposes without treatment on account of the presence of sewage which will cause foaming, or of certain salts which will form

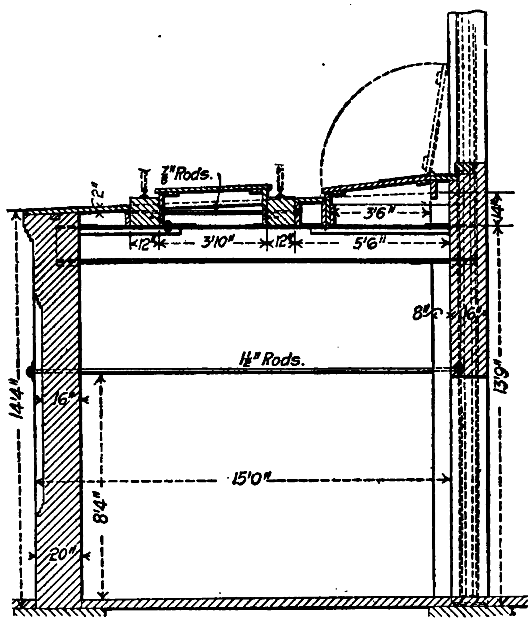


Fig. 66. Coal Bunkers designed by Sheaff and Jaasted.

hard scale on the heating surface of the boilers and not only impair their efficiency but also entail large expense for cleaning them and be a source of probable danger besides. Deep well waters frequently contain scale-forming salts and occasionally rivers receiving the rainfall from certain watersheds. River water is frequently contaminated with sewage. Sewage can probably be best removed for boiler purposes by mechanical filtration; silt and mud by sedimentation and the same process.

**Water Softening.** — The salts which give the most trouble in steam-boiler waters by the formation of scale are the carbonates and sulphates of lime and magnesia. The presence in water of these salts, except the sulphate of magnesia, cause it to be hard,

and their removal is known as water softening. Sulphate of magnesia does not cause scale to form, but it is precipitated by concentration. Its presence prevents the removal of other salts by chemical treatment due to the fact that when lime water is added to water containing it, it breaks up and sulphate of lime results, which does form scale. A small quantity of carbonate

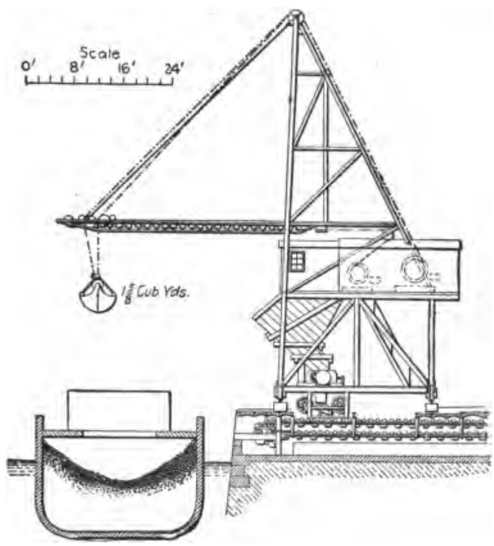



Fig. 67. Typical Coal Elevating Tower.

of lime is apt to remain in solution in almost pure water, but if the water be saturated with carbonic acid the amount of carbonate of lime in solution can be very much greater; most of it being precipitated when the carbonic acid is driven off, as it is when the water is heated. By adding lime water to water containing carbonate of lime and carbonic acid, the lime combines with the carbonic acid to form carbonate of lime which, with the carbonate of lime previously in the water, is precipitated because of the disappearance of the carbonic acid. Carbonate of magnesia is very similar in its properties to carbonate of lime. It is also acted upon by the lime water so as to form hydrate of magnesia, an almost insoluble salt, and carbonate of lime which is precipitated. The removal of the carbonates by the addition of lime water was proposed by an English chemist named Clark and this process generally bears his name.

Sulphate of lime and sulphate of magnesia may be broken up by adding sodium carbonate, the former resulting in the formation of calcium carbonate, which is precipitated, and sodium sulphate, a nonscale-forming salt. The magnesium sulphate is split up into sodium sulphate and carbonate of magnesia and the latter can be further broken up by adding lime to cause an additional reaction in the manner described. The amount of sulphate of lime that can be dissolved in water depends upon the temperature, but above a temperature of about 100° F. the solubility of this salt diminishes. At 300° F. it is said to be insoluble. By heating with live steam, therefore, it is possible to precipitate nearly all of the sulphate of lime in water containing it. Quite a little time, however, is required to complete the reaction. Heating water carrying sulphate of lime previous to its entering a boiler will only remove the excess of sulphate of lime over that possible to retain in solution at the temperature to which the water is heated, so that unless it is heated to a high temperature chemical treatment is necessary to precipitate that not precipitated by heating.

It will be noticed from what has been said that the carbonates of lime and magnesia can be mostly removed by heating, particularly if sufficient boiling occurs to drive off the carbonic acid, and some of the sulphate of lime can also be precipitated by heating. All of the four salts mentioned can be practically removed by chemical treatment. The heating method is more limited in its application but it is cheaper when it can be used. In certain cases chemical treatment or a combination of chemical treatment with the application of heat is alone possible. The best method can only be determined by a competent chemist after a proper investigation of the water is made, as it depends entirely upon the kind and quantity of salts present in the water.

**Purifying with Exhaust Steam.** — Almost all of the carbonates of lime and magnesia can be precipitated by moderate warming such as could be accomplished by exhaust steam in a feed-water heater of the open type. These are usually provided with trays over which the water trickles and filtering material to intercept the precipitated salts that do not lodge on the trays. Care should be taken that the heater is large enough to insure a low velocity of water passing through it to give time for the precipitation to occur in the heater and not in the boiler. Very often

boiler waters are only objectionable because of the presence of salts which could be entirely removed practically in an exhaust-steam heater.

**Purifying by Live Steam.** — A live-steam purifier is a device similar in a general way to most open-type feed-water heaters in that it consists of a shell containing a number of trays that can be withdrawn by removing the head of the heater, the trays being arranged one over the other so that the water trickles slowly downward over them so as to become thoroughly heated by the live steam under full boiler pressure that is admitted to the heater. The heater is usually located over the boilers so that water will run from it to the boilers by gravity. As live steam is used in these heaters and the temperature is higher, much more of the sulphate of lime present in water will be precipitated than there will in an exhaust-steam heater. This type of heater will eliminate the carbonates of lime and magnesia.

**Purifying by Chemical Treatment.** — The chemical treatment might be divided into two methods, the intermittent and continuous systems, and perhaps a third, with either combined with the application of heat to the water while being treated for the purpose of hastening the action of the reagents. The original Clark process was the intermittent method and it has been modified more or less by others. Usually this system consists of two large settling tanks in which the water to be treated is run, the proper amount of chemicals added, the mixture agitated and then allowed to stand for some hours while the precipitate settles, the clear treated water on top being used and the sludge that settles being blown off when necessary. This system usually requires two tanks so that the treatment can go on in one while purified water is being drawn from the other. An objection to it is the first cost. With the continuous method the untreated or raw water and the necessary chemicals are mixed in the desired proportions, the supply of both being relatively constant, both being fed into a tank of generous size so arranged that the current is sufficiently slow for the precipitate to settle in the bottom of the tank, the clear water passing on to the outlet. The continuous treatment combined with heat has been used with no little success. The Sorge-Cochrane system of water purification consists in heating water almost to the boiling point, purifying by a simple chemical treatment and at the same time

neutralizing such acids as are present in the water. The supply of water to the boilers is made up of all the pure condensation that can be saved and utilized, and of just enough fresh cold water supplementing this condensation to supply the demand. The conversion of the sulphates and carbonates of lime and other soluble salts into insoluble and neutral salts is accomplished in a specially designed feed-water heater of the open type by introducing into the water to be tested, before it enters the heater, a suitable chemical such as soda ash and simultaneously heating the water. Means are provided for varying the amount of the reagent, and also, by means of a filter bed in the heater, of intercepting the precipitated salts and other insoluble matter.

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